PROCEEDINGS

3rd National Symposium

on

Refrigeration

and

Airconditioning

18-20 July 1974

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FOREWORD

In a tropical country like India with varying conditions of weather, pockets and seasons of glut and scarcity, refrigeration has an important role to play in the preservation of perishable foods such as fruits, vegetables, meat, fish, milk and other dairy products.

The process of deterioration is slowed down considerably by the application of refrigeration and thus the food is saved from loss. Although the first cold storage was established as early as 1892, the growth of the cold storage industry has been rather slow for a number of years. But, of late, it has gained momentum. Yet it is nowhere close to meeting the needs. Even though statistics show that the number of cold storages in the country has increased 17 times, while their capacity has increased 20 times during the past 15–20 years, this growth is just not adequate to offer even a minimal degree of relief to the problems of the food preservation industry. Nearly 80 per cent of the 1400 cold storages in the country today are used for storage of potatoes only. The remaining cold storages are used for fish and dairy products. A very small capacity is utilised for the storage of fruits and vegetables at present. The organisation and development of cold storage industry, therefore, on sound lines is not only an economic necessity but is also important for curtailing the losses of the highly perishable commodities.

The frozen food industry in India is still in its infancy. At present only fish is frozen and most of it is exported. With the present cost of traditional packaging, the prospects for frozen foods in the Indian market appear to be quite bright. New techniques and equipment for handling and freezing of various food commodities need to be developed and made available on reasonable prices so that the refrigeration industry becomes self-reliant.

Air-conditioning and environmental control in India is no longer considered a luxury. In the context of present day energy crisis, optimal thermal design of buildings and air-conditioning systems have become a necessity in order to conserve the energy. Use of solar energy in refrigeration and air-conditioning requires first of all adequate inputs of research and development. The future prospects for such a development are very bright.

The 3rd National Symposium on Refrigeration and Air-conditioning held at CFTRI in July 1974 has brought into focus the problems of the industry and the areas where development is urgently needed. Among other things the Symposium has discussed the latest researches and techniques, designs and development in Refrigeration and Air-conditioning. Scientists, technologists, manufacturers of machinery and equipment and users have contributed useful papers based on their work and experience in the related fields. The Symposium of this topic has now become an annual feature and provides an opportunity to all the participants to exchange information and ideas on this vital subject.

A total of 40 papers were presented in the 5 technical sessions. The papers have been edited and published for the information of all concerned. The document also includes the conclusions and recommendations made during the Symposium for follow-up and implementation for the development of Refrigeration and Air-conditioning industry in the country.

On behalf of the Organising Committee, I wish to express our sincere thanks to the Council of Scientific and Industrial Research, Ministry of Defence and other government organisations, research institutions, industrial concerns and all others who extended their full co-operation and assistance in organisation and success of the Symposium.

B. L. AMLA
Organising Committee Chairman

Welcome Speech

B. L. Amla

Dr Ramachandran, Mr Balaraj, distinguished guests, participants to the Symposium, ladies and gentlemen :

It is a privilege and pleasure indeed for me to welcome you all today on behalf of the Organising Committee to the inauguration of the 3rd National Symposium on Refrigeration and Air-Conditioning.

We are very fortunate in having Dr A. Ramachandran, Secretary, Department of Science and Technology, Government of India to inaugurate the Symposium. Dr. Ramachandran as you all know is a pioneer in the field of refrigeration and air-conditioning. Before moving to Delhi, he has been the Director of Indian Institute of Technology, Madras where he started the R & D work on refrigeration and air-conditioning. Under his able guidance a scheme for establishing a centre of cryogenic is being established at IIT, Madras. His efforts in this direction will have a long reaching effect on R & D necessary in the processing, preservation and transportation of highly perishable foods. We are indeed grateful to him for responding to our request to inaugurate the Symposium and deliver the key note address in spite of his busy schedule of engagements in Delhi. Those of you who are well aware of the happenings in Delhi would agree with me that the assignment Dr. Ramachandran is holding is always taxing on ones time. I, on behalf of the Organising Committee and the participants extend him a special welcome.

We are very happy to have with us Mr. Balaraj, Development Commissioner, Government of Karnataka, who has so kindly consented to preside over this function. He is no stranger to Mysore as well as to CFTRI. Before moving to Bangalore, Mr. Balaraj has been the Divisional Commissioner in Mysore. In his present assignment as Development Commissioner he is directly connected with the planning and production of agricultural commodities and their post-harvest handling, storage and processing. We are indeed grateful to him for having agreed to come to Mysore with all the multifarious engagements he had for the day. As a matter of fact, he is just coming out of a meeting from Bangalore. Please allow me to extend you a hearty welcome.

It was on the recommendation of the Indian National Committee of the International Institute of Refrigeration made at its last Symposium held in University of Roorkee in 1973 that the 3rd Symposium should be held at Mysore and CFTRI should take up the responsibility of organising the same in collaboration and association with Union Ministries of Defence, Agriculture, All India Air-conditioning and Refrigeration Association, Institution of Engineers, India, Mysore City Sub-Centre and Association of Food Scientists and Technologists (India). It is, therefore a joint effort of all the organisations interested in the promotion of R & D efforts in the area

of refrigeration and air-conditioning. The Symposium will bring into focus the problems faced by this industry and identify the areas requiring further development. During the deliberations of the Symposium the latest research and technology design and development in the field of refrigeration and air-conditioning will be discussed at a greater length. The Symposium will provide a forum for the scientists, technologists, industries engaged in the manufacture of machinery and equipment and the users, to exchange information on the newer development and also pose current problems faced by them in their endeavour. It is therefore our hope and expectation that the conclusions and recommendations which may emerge at the end of the Symposium will be followed up and implemented for the development of refrigeration and air-conditioning industry in the country.

There are nearly 180 delegates including design engineers, research scientists, manufacturers, management executives and government representatives attending this Symposium. Such a galaxy of personalities only shows the importance of the subject in the context of country's requirements. A wide spectrum of papers from specialists will make the technical sessions a forum for lively discussions and exchange of ideas.

On behalf of the Organising Committee and on my own behalf, I extend to all of you a hearty welcome again. I hope you will enjoy your stay at Mysore. In case you experience any inconvenience, please bring it to the notice of the organisers and the volunteers who will be too happy to assist you.

With these words, I wish you all a very happy and fruitful deliberations during the Symposium.

Thank you.

INAUGURAL ADDRESS

Heat and Mass Transfer Problems in Food Processing and Production

A. RAMACHANDRAN

Food products, whether of animal or plant origin, undergo many processing treatments before they are consumed. These operations may be performed to preserve the product against deterioration, or to reshape the product into suitable form for ease of handling, transport and storage, or to obtain new products. Also, the processing operation may be purely mechanical, chemical, thermal, or may be any combination of these. However, in a majority of cases, food products undergo thermal treatments (heating and/or cooling) at one stage or another. The most common operations are blanching (including scalding and cooking), pre-cooling, freezing and drying.

Food products may be processed in solid or liquid state. Fruits, vegetables, meats, fish, poultry, etc. are solid foods, whereas most of the dairy products, fruit juices, etc. come under liquid or semi-liquid (slurries and pastes) category.

Solid foods are porous bodies with high moisture content. Hence, any thermal treatment is accompanied by a moisture migration within the body and also moisture transfer at the surface to the ambient. Compared to other porous bodies like leather, textile, timber, building materials, insulating materials, etc., heat and mass transfer in food products is a more complicated problem because, rigorous consideration has to be given to the retention of quality (appearance, colour, flavour, structure, texture, vitamin content, etc.). This imposes severe restrictions on the processing operations.

Properties of food products

In order to study the heat and masstransfer in food products, it is essential to know the thermo-physical and transport properties. The important ones are specific heat, thermal conductivity, thermal diffusivity, density, critical moisture content and moisture diffusivity. There is a scarcity of property data, particularly of those related to moisture transport. The dependence of these properties on temperature and moisture content are important. However, in many processes, the temperature differentials are kept low and hence temperature dependence may be insignificant. But the dependence on moisture content is important because in many dehydration processes, the moisture content changes from as high value as 90 per cent to as low a value as 5 per cent. Properties at low temperatures are important, particularly for freezing and cold storage calculation. Another compli-

cating factor is that the properties under thawed conditions are many orders of magnitude different from those under frozen conditions. Presently, data on densities and specific heats of food products are available in tabulated form. Attempts have been made to collect together the thermal conductivity values. The data regarding the moisture transport properties are rare. It is important to note that the same variety of products may have different property values under different conditions. The property values of products of plant origin are affected by the locality, nature of soil, season, fertilizers used, harvesting methods, storage conditions, etc. The properties of products of animal origin differ from animal to animal depending on its food habits, environment, breed, physical conditions, etc. Another factor which influence the properties is the structure of the product. Most of the fruits and vegetables are cellular in structure and in absence of seeds and outer cover (which are usually discarded before processing) may be considered to be homogeneous. But most of the meats are fibrous in nature and have properties which differ very greatly from across the fibres to along them. Most of the available property values have been determined under steady conditions. These may vary greatly under the dynamic processing conditions. The movement of moisture, natural convention, and radiation effects, phase change, etc., may significantly influence the effective property values during actual processing. A knowledge of properties in vacuum or in the presence of gasses other than air, are of importance for studying processes like freeze-drying, vacuum drying, vacuum cooling, etc. Most of the property data that are tabulated are for food products that are produced and consumed in western countries. For instance, the properties of

Dr A. Ramachandran delivering the inaugural address

tropical fruits like mango, banana, pineapple, guava, etc., are not definitely known. Also, the properties of most of the food grains and spices are unknown. Though a number of theoretical models are available to represent the thermal conductivities of porous bodies, their use for the case of food products is limited as they require a detailed knowledge of porosity, solid conductivity, and above all, the structure of the product. Hence, experimental determination seems to be the only possible solution. In conclusion, one may state that it is extremely essential to determine, collect and tabulate the property data in a systematised pattern. This is important to have a control over the economy of the processes and the quality of the products.

Cooling

Cooling of food products to retard the deterioration has been a long known method. The two most common processes are 'precooling', prior to transportation and distribution, and 'freezing', to achieve increased storage life.

Aircooling, hydrocooling, cooling in brine or sugar solutions and vacuum cooling are some of the precooling methods commonly used. Determination of the cooling leads and time-temperature characteristics which is important to exercise a control over the quality of the product and on the economy of the process. Presently, conventional Heissler's and Curney-Lurie charts are used for determining the heat transfer during the cooling process. This introduces substantial errors (particularly in aircooling calculations), because the heat transfer at the product surface is not one of pure convection. A moisture film usually exists at the food surface due to the usual 'washing' treatments. The high moisture condition of the food products also helps to sustain the moisture film due to migration of free water towards the surface. The coupled heat and mass transfer effects due to this constantly evaporating moisture film has a significant influence on the total energy exchange. This necessitates the use of separate charts for evaluating the heat transfer characteristics. These have recently been developed at the Refrigeration and Airconditioning Laboratory of Indian Institute of Technology, Madras. Also, consideration has not been given to the prediction of vacuum cooling characteristics. This deserves greater attention since this happens to be one of the most efficient methods of cooling the thin and leafy products. Experimental investigations on precooling of food products are reported in plenty. But most of them have been conducted on specific food products under particular processing conditions. Generalised studies depicting the influence of various processing parameters are scarce.

Freezing

Freezing of food products in order to prolong the storage life has been one of the earliest methods. Various aspects of freezing such as, nucleation, ice crystal growth, freezing rates, effects of freezing rates on the product structure, freezing injury, freezer burn, drip losses, etc., have been studied in detail both experimentally and analytically. The basic mechanism of freezing seems to be well understood. However, there is a necessity for mathematical models for predicting the freezing characteristics of anamolous shapes. Concentrated efforts are needed to economise the comparatively new techniques of freezing with LN2, LF 12 and LF22 spray. It is

important to design the equipment for maximum heat transfer with minimum wastage of the cooling medium. Also, product damage due to thermal shock (often resulting in 'cracking' at the product surface) needs to be taken care of. Freezing still remains to be a major mode of food preservation in spite of the fact that a 'cold chain' has to be maintained from the place of production to the place of consumption. However, efficient methods of dehydration are being sought after to break this cold chain.

Dehydration is the reversible removal of water from the food product. The important considerations are the economy of the process and the quality retention of the product upon rehydration. In most cases, (particularly for solid foods) air is the heat and mass transfer media. Air-drying is carried out in tunnels, cabinets, shelfs, bins, fluidized beds, particle beds, etc. For liquid foods, drum drying, foam mat drying, vacuum puff drying and spray drying are adopted. Recently, vacuum freeze-drying is gaining popularity for dehydration of both solid and liquid foods because it yields very high quality products. However, the exhorbitant initial, maintenance and processing costs prevent the universal application of the freeze-drying process.

Drying of porous bodies has been a vastly studied subject because a number of engineering materials fall into this category. However, most of these studies are applicable for the vapour phase transport of moisture i e. to the later stages of falling rate period. Food products contain high moistures and due to quality retention considerations, are processed under not too intense conditions. Under these 'low intensity' and 'high moisture' conditions, the moisture migration within the body may be considered to be governed by 'liquid diffusion' and the moisture loss at the product surface to the ambient may be assumed to be controlled by 'surface evaporation'. Convective drying under these conditions is of particular importance for food products. Hence, theoretical models and experimental investigations applicable to low intensity and high moisture conditions are needed.

Important aspects in economizing the freeze-drying process are, the improvement in heat transfer from the heater to the food surface and from the food surface to the subliming ice front, effective removal of sublimed water vapour and the reduction of the drying time. To achieve these goals, a number of techniques have been suggested such as spiked plate freeze-drying, accelerated freeze-drying (A.F.D), introduction of inert gases like hydrogen, helium, nitrogen, carbon dioxide, etc., into the drying chamber. Since it is uneconomical to maintain high vacuum, the recent trend is to operate at not too high vacuum. This introduces the possibility of gas conduction and convection efforts being significant in addition to radiation heat exchange between the heater and the food surface. This aspect needs detailed investigation and suitable correlations for the heat transfer at the product surface to help the process and the product designer in predicting the freeze-drying characteristics. Reduction in the product size is an efficient method to reduce the processing time. Hence, the recent trend is to freeze-dry pebbles or granules in fluidized or particle beds. These techniques require detailed analytical and experimental investigations.

Spray drying is a universally accepted method of dehydration of liquid foods. The various aspects of spray drying such as spray nozzle design, atomization, air flow pattern in the chamber, etc., have been studied in detail and the spray drier

design criteria is well established. However, vacuum puff drying, foam mat drying and freeze-drying yield products of better quality. But spray drying wins over all these methods because of its simplicity, economy, versatality and smaller drying times. Rigorous studies on heat and mass transfer mechanisms during the other methods of drying are essential in order to make them economically comparable to spray drying. A technique recently being tried is sprary freeze-drying wherein the advantages of both spray drying and freeze-drying are incorporated. The liquid freezes when sprayed into an evacuated chamber and deposits on the drier wall where the heat of sublimation is supplied. This seems to be a very promising method and deserves greater attention.

Solar processing of food

Intensive studies on the efficient and economic conversion of solar energy into useful 'heat' or 'cold' have led to the development of solar water heaters, boilers, air heaters and refrigerators. Food processing is a promising field wherein these equipments can be advantageously used. This is particularly applicable for tropical countries where sun's energy can be harnessed in abundance. Solar energy is of particular utility for food processing because, the farms, animal sheds, barns, and the processing factories usually occupy large areas exposed to sun's radiation. In these cases, simple, blackened, corrugated sheet roofs can be used as solar collectors. Most of the food grains and other exclusively tropical products like coffee, tea, pepper, cardamom, copra, etc., are invariably dried before use. Indirect solar heating has been found to yield products of better quality than those directly sun-dried.

Another possibility of utilizing solar energy is to provide the refrigeration for cold stores. The major heat load on a cold store being the wall heat gain, if the roof and the walls facing east and west are used for tapping solar energy, it will also help in reducing the wall heat load. This heat can be used to operate the absorption refrigeration systems. It is interesting to note that maximum solar energy is available when cooling inside the cold store is the most needed.

Other areas of food processing where solar energy can be advantageously employed are in the evaporation and concentration of liquid foods, generation of steam for blanching, cooking and peeling, and for creating vacuum (using steam ejectors) for many operations such as vacuum cooling, vacuum puff drying or freeze-drying. Hence food processing applications of solar energy makes an important field of study, particularly for tropical countries.

Presidential Address

D. J. Balaraj

I am happy to be with you this evening presiding over the inaugural function of the Third National Symposium on Refrigeration and Air-conditioning. The fact that the organisers have thought of bringing a practitioner of planning and development like myself to this function where a large number of technical experts, industrialists and others have assembled for discussing the problems and prospects of the refrigeration and air-conditioning industry shows the high appreciation and utility of exchange of ideas between the technical experts and the planners for achieving the objectives of planning and for giving the required orientation to the programmes of this industry as such. I am thankful to Dr B. L. Amla and you all for this gesture and opportunity.

You have chosen Mysore City as the venue for the Symposium. This is significant for more than one reason. The obvious reason that any strike you immediately is that C.F.T.R.I. has done a great deal of work on the preservation of fruits and vegetables by refrigeration and therefore C.F.T.R.I. located at Mysore is the logical choice. The other reason which I would like to mention is that, in a way, Mysore City is nature's air-conditioned city and it is but appropriate that a Symposium on Refrigeration and Air-conditioning should itself be held at an 'air-conditioned city'.

You are brightly concerned with the present status of refrigeration and air-conditioning industry in the country and the projections and proposals for its future development. On the technical aspect of the industry, I do not wish to make any pronouncements since I am not an expert in that field. As for the problems and perspective, I do have some views which I am glad to share with you this evening with the hope that they may receive some consideration during your deliberations.

Broadly, refrigeration and air-conditioning impinge on both agricultural and industrial policies and programmes. On the side of agriculture, the national goals are (1) to increase the production of foods rich in proteins either of plant or of animal origin or better still a judicious combination of both ; (2) to increase the production of foods rich in minerals and vitamins like fruits and vegetables ; (3) to increase the production of non-food crops including various products so necessary to feed the industries to meet the increasing internal demands and also for export to earn valuable foreign exchange ; (4) to continuously increase and stabilise production at such a level that it would be sufficient to meet the internal demands even in lean years and surplus for export to exportable products. The refrigeration and air-conditioning industry is very vital to goals (1) and (2) and has also an important role to play in respect of goals (3) and (4).

On the side of industry, careful husbanding of our resources in the context of the core sector and proper priorities will receive greater thought. Being an export

item, refrigerators and air conditioners should have a high target and quality performance. Installed capacity is to be fully utilised to reduce costs and maximise exports. There is also an increasing domestic market. More than this, infrastructural facilities like storage, transportation, health and the requirements of many industrial operations involving precision put their claims on this industry.

With shortages in food production, it is imperative to eliminate any wastage that may occur in the production of fruits, vegetables, dairy, poultry and fish products and in storage of grains It is estimated that out of annual production of fruits and vegetables alone, nearly 20 to 25 per cent raw materials go waste because of improper packaging, transport and marketing facilities which is a handicap both to the producers and consumers. The problem of preservation of perishable food like fruits, meat, fish, eggs, milk and milk products, fats and oils, starch roots, etc. has assumed a new dimensions in our country. At present the consumption of various perishable commodities form nearly 36 per cent of the total food consumption as against 78 per cent in developed countries. It is therefore evident that refrigeration is needed to preserve the perishable products. The refrigeration industry can play a vital role in eliminating the wastage to a great extent.

It may be relevant at this stage to refer to the magnitude of the problem of refrigeration in my own State, Karnataka. The major perishable food articles like fruits and vegetables, dairy products and fish products play a significant role in Karnataka's economy. The total area under fruit crops in the State is about 2,75,010 acres. Mango is cultivated in 59,696 acres, banana in 98,625 acres, citrus varieties in 57,546 acres, gauva in 9,785 acres, sapota in 8,595 acres, grapes in 10,357 acres, pineapple in 2,634 acres, papaya in 9,350 acres, potato in 47,685 acres, tomato in 19,721 acres and peas in 4,685 acres. The estimated production in some of the major fruit crops is about 1,20,000 tonnes of mango, about 9,00,000 tonnes of banana, 20,000 tonnes of pineapple, 3,80,800 tonnes of potato, 1,19,820 tonnes of tomato and 1,75,000 tonnes of citrus fruits. The production of fish is estimated at 1,90,000 tonnes, of prawns 50 million tonnes and of fry and fingerlings 12 million tonnes.

Government have provided processing facilities for the preservation of fruits and vegetables in 44 units. At important fish landing centres, Government have set up a number of ice plants and cold storage and freezing units.

There are 9 milk plants with capacity per day of 1,17,000 litres. The Fifth Plan aims to raise the number of plants to 11 and the capacity to 6,70,000 litres. The World Bank has sanctioned a Rs. 50 crore Animal Husbandry Project for the State which will be implemented over a period of 7 years. Frozen Semen Centres, chilling plants and dairy products create a huge demand for the air-conditioning and refrigeration equipment.

Thus, the State has already been making use of the technology in many of its developmental programmes. However, when compared with the potential of the state in fruits, vegetables, fishing, animal husbandry and poultry, there is still a great leeway to be made up in this state, as in our country, in this field.

It is not enough if more food is produced and stored or preserved at certain centres which have natural endowments. Food thus produced will have to be carried to the different parts of the country in time. This is a herculian task in a big

country like ours. Proper distribution would call for appropriate collecting, grading, prepacking, cold storage and transportation measures. In their absence, no wonder the prices of fruits, vegetables and many of the other perishable products vary very much between metropolitan cities and smaller towns.

For reducing high prices, especially in some smaller towns and for ensuring proper distribution of food items like milk, eggs, meat, potatoes, fish, mangoes, bananas, etc. cold storage facilities will have to be built at centres which ensure a fair spatial distribution. This raises two questions: one relates to the optimum size for small centres and the other relates to its cost. Experts who are assembled here will, I believe, find answers to these questions. Perhaps, these have already engaged attention of the food scientists and technologists at the Seminars and Symposiums held in recent years like the International Seminar on Refrigeration Applications to Fish, Fruits and Vegetables held at Durgapur in January 1974 and the Symposium of Food Scientists and Technologists held in Calcutta in March 1973. The point which I am trying to stress is that your deliberations must try to relate the technological solutions, which you may recommend for solving the problems of this industry, to the objectives of lowering the cost and planning for spatial distribution of cold storage facilities. Also, they must help minimising poverty by providing employment opportunities on programmes of this industry which support the developing of milk and dairy industry, fruit preservation industry, fish products, marketing, etc.

Cold storage for food is only one important application of refrigeration. There are other applications which are equally significant. I may mention chilled water which is required for curing the heavy concrete masses used in concrete dams and for cooling reaction vessels in the pharmaceutical and chemical industries. In a larger number of industrial operations like optical goods, laboratory apparatus, drugs, jet engines for aircraft, computers, control rooms, production of precision machinery etc. I believe the applications of air-conditioning are increasingly found. Besides, air-conditioning plays a life saving role in hospitals especially in operation theatres, intensive care wards etc. Of course air-conditioning is being increasingly resorted to in buildings particularly in cities which have a hot climate. Even otherwise, affluence creates a need for it. Even Mysore, despite its salubrious climate, may be no exception.

With such increasing uses of air-conditioning and refrigeration, it may be incredible to be told that the capacity utilisation in the case of domestic refrigerators is about 63 per cent and the industrial air-conditioning and air control equipment about 46 per cent in 1973 in our country. Whatever the reasons, two implications follow from this. One is that heavy investment has been made in this line which is somewhat capital intensive and resources are made idle. The second is connected with the composition of output and the focus we should accord for the future development of this industry.

It is likely that eye brows will be raised if I say that air-conditioning and air control equipment are being used even in cities whose climatic conditions do not demand their uses and an indiscriminate adoption is only an indication of the wastage of scarce resources. Refrigerators all looked upon more as symbols of status than of necessity. I therefore feel that from the view point of getting the maximum benefits

from our scarce resources, we have to apply more rigorously the tests of priority in this field.

It is gratifying that the Indian manufacturers of refrigerators and air-conditioning equipment are increasing the indigenous content of their products and are even making new items with the help of research and development established in the country. They have also managed to secure some large orders for exports. This is however not sufficient. Having established a large capacity, the manufacturers have to work hard for maintaining quality and capture more of the foreign markets by offering competitive prices.

It may be maintained that air-conditioning and refrigeration industry may be allowed to capture a larger domestic market by a reduction in the excise duty and sales tax. In fact one hears of the complaint voiced by this industry that the excise duty and sales tax are very high and they are a disincentive to increase production. Therefore, a suggestion has been made that these taxes should be reduced so that the industry will get a fillip to expand production and capture a large proportion of the domestic market. The argument is sound in so far as it goes. But when subjected to scrutiny in the light of the problems of the country and the socio-economic objectives of planning, I am afraid this line of thought has to be changed. We are using our scarce resources for catering to the needs of the elite, the top ten per cent of the population, in this country. Bank finance is also attracted by these items because they become commercially viable whereas essential projects like those of power starve for want of funds. There is something wrong in our sense of priorities. We should aim at achieving a pattern of production in the refrigeration and air-conditioning industry which will help catering to the requirements of the masses. This can be done by meeting the processing and storage facilities on a large scale to promote the development of the fish industry, animal husbandry and dairy programmes, fruit and vegetable preservation, supply of hybrid seeds, etc. In my view, what is needed is not a reduction in the excise duty or sales tax but a deliberate attempt to adapt and use the capacity we have in this industry to the programmes deriving support from the masses.

Significant efforts have, no doubt, been made in recent years through research and development to carry the frozen food industry a step forward. Mostly this work has been done by the C.F.T.R.I. or its subsidiaries. I also understand that the I.I.T., Madras and more recently a firm in Delhi have made some contributions in designing contact plate freezers. Side by side research is also going on storage and transport of chilled or frozen foods, on fluidized compact freezing and on evaporative cooling equipment for post-harvest use. While these efforts are commendable, the energy crisis confronting the world as a whole today poses new challenges to the designers of machinery. The existing resources of fuels and hydro-electric power have to be sparingly and most profitably used. Research is necessary in further economising the use of power in air-conditioning and refrigeration plants. One may have to balance the costs of the new techniques against the benefits more carefully before accepting or rejecting the commercial application of the results of research and develompent.

I referred earlier to what the Karnataka Government have done in the matter of providing cold storage and other processing facilities for the preservation of fruits

and vegetables, animal husbandry and dairy products, fish products etc. As more and more units are established in different centres, there arises the problem of maintenance and repairs. It may so happen that a costly refrigeration or air-conditioning plant will have to go idle for want of timely technical services for repairs. This will mean another form of wastage of capital resources affecting output and will have to be arrested. We can tackle this problem, if we develop technical man power both for their maintenance, repairs as well as for further expansion. At the various polytechnics and other training centres, the requirements of skills relating to air-conditioning and refrigeration have to be given adequate attention by suitable framing of syllabus and curriculum. More than this, I strongly feel that the air-conditioning and refrigeration industry must participate in the training programmes in a big way. It is only when such close co-ordination obtains between the industry and the training institutions that our educational system will get work orientation and the maladjustment we notice between the out-turn of the training institutions, especially the polytechnics and industrial training centres, and the employment opportunities for them can be minimised and eliminated. It is not enough if the industry participates in the drawing up of the curriculum. It should participate in offering courses and in providing in-service training. Even the finances needed for such training should come from the air-conditioning and refrigeration industry.

I have placed before you some of the thoughts which were uppermost in my mind. I may or may not have succeeded in convincing you about them. When scientists, technologists, Industrialists and other experts have gathered, I consider such an occasion as an appropriate one for enlarging the horizons of discussion even

Presidential address by Shri D. J. Balaraj

if there were to be any disagreements. The 'agitation' to your intellect, which I may be causing is surely no violence judged in the context of scores of other serious agitations which we witness these days. Without new challenges, no progress is possible. I have only tried to do the simple task of drawing the attention of the air conditioning and refrigeration industry to the challenge of the new social order which is being ushered in this country through planned development.

With these observations, may I wish the Symposium every success ?

Shri S. K. Lakshminarayana proposing the vote of thanks

L to R : Dr T. N. Ramachandra Rao, Dr H. Nath, Dr A. Ramachandran, Mr D. J. Balaraj, Dr B. L. Amla, Dr A. K. Dey

Delegates of the Symposium

Another view of the Symposium

A view of the exhibition arranged in connection with the Symposium

Another view of the exhibition

SESSION I

Cold Storage of Perishable Products

Chairman
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Rapporteurs
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Cold Storage of Grapefruit
(Citrus paradasi MACF*)*

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In India, grapefruit is not considered as important a crop as mandarins or sweet oranges. However, presently its popularity is on the rise on account of its preference by foreign visitors. Grapefruits are available in the market for one or two months and in order to make this fruit available during off seasons, it is necessary to extend its storage life at low temperature. The storage behaviour and subsequent chemical and physical changes in the fruits during cold storage have been reported by workers in other countries.[1-8] The major disorder with grapefruit is pitting of the rind at 31-42°F.[1,2] Climatic conditions and soil also influence the storage disorder in grapefruits.[5] For decay resistant fruits, 45-55°F storage temperature was found to be suitable, however, those which were susceptible to decay required 37.5 to 42°F.[2,4]

As very little information is available on the storage behaviour of different varieties of grapefruits grown in India, studies were undertaken to determine the optimum cold storage conditions for three varieties of grapefruits namely *Saharanpur Special, Marsh Seedless* and *Ruby*.

Materials and Methods

Three varieties, *Saharanpur Special, Marsh Seedless* and *Ruby* were obtained from the orchards located in the Horticultural Research Institute, Saharanpur, U.P. Fully matured grapefruits were harvested in the month of December. These fruits were washed in water with 2 per cent hypochlorite for a period of two minutes to remove dirt and for surface sterilization. A lot of 50 fruits of each variety was packed in wooden crate with paper lining and stored at low temperature, initially, the storage studies of variety *Saharanpur Special* were carried out at six different temperatures, 32-34, 35-37, 40-42, 40-43, 45-48 and 48-51°F. This variety grown in two seasons was compared for its storage behaviour and quality with other two varieties, *Marsh Seedless* and *Ruby* at 48-51°F. Periodically, five fruits were picked from each treatment and analysed for total soluble solids (TSS), juice percentage, acidity and vitamin C contents. The values during storage were compared with the initial freshly harvested values of the fruits. Total soluble solids in juice were determined by hand refractometer. Acidity was determined by titrating aliquot portion of the juice by standard alkali and calculated as citric acid. Vitamin C was estimated by visual dye titration method using indophenol dichlorophenol.

Results and Discussion

Results are given in Tables 1, 2 and 3. From Table 1 it is seen severe pitting

TABLE 1. *General condition of grape fruits variety* Saharanpur Special *stored under different temperatures*

1st Set			2nd Set	
Tempertaure of storage °F	General condition and pitting		General condition and pittings	
	7 days storage	18 days storage	Temperature of storage °F	1 month storage
32–34°	Very good no pittings	Bad severe pittings	40–43°	Worst pitting severe
35–37°	-do-	-do-	45–48°	Good pitting nominal
40–42°	Less glossy no pittings	-do-	48–51°	Good pitting nil

TABLE 2. *Chemical changes in grape fruits variety* Saharanpur Special *stored under different temperatures*

Temperature of storage °F	Juice %		T. S. S. %	Acidity as citric acid %	Ascorbic acid mg./100 g.
	W/W	W/V			
Initial	46.3	45.6	10.0	1.32	35.3
After 4 weeks of storage					
40–43°	46.8	45.0	10.5	1.76	34.7
45–48°	48.4	46.1	10.5	2.02	34.7
48–51°	47.8	46.4	10.0	1.65	38.2
After 10 weeks of storage					
40–43°	47.5	46.2	11.0	1.34	35.2
45–48°	53.3	49.5	10.0	1.27	35.5
48–51°	47.6	46.4	9.5	1.19	36.3

in the rind occurred 18 days after storage at temperature 32-43°F. At 45-48°F the pitting was less severe as compared to those stored at 48-51°F which showed no pitting after one month of storage. The general condition of the fruits at 48-51°F was the best as they were free from any disorder such as rind pitting and discolouration. The changes in the chemical constituents of the fruits during storage at all temperatures were not significant. However, acidity in fruits gradually decreased as the storage period advanced at low temperature of 48-51°F.

From Table 3 it is clear that three varieties *Saharanpur Special, Marsh Seedless* and *Ruby* when stored at 48-51°F for five months were free from rind pitting, discolouration and soft texture. Further storage at this temperature promoted soft texture and poor juicy quality.

From the foregoing, it is concluded that three grapefruit varieties—*Saharanpur Special*, *Marsh Seedless* and *Ruby* can be successfully stored at 48-51°F for a period of five months without any disorder such as rind pitting and with minimum of changes in juice and vitamin C contents and total soluble solids.

Table 3. *Periodical analysis of grape fruits of different varieties stored at 48–51°F.*

Months	Juice %	T.S.S. %	Acidity as critic acid %	Ascorbic acid mg/100 g	General conditions
			Ist YEAR		
Saharanpur Special					
0	33.4	9.0	2.66	38.03	No pittings, good
1	32.4	10.0	2.14	41.11	— do —
2	36.0	9.0	1.95	31.18	— do —
3	34.5	9.0	1.47	43.75	— do —
4	33.0	7.0	1.02	41.50	— do —
Marsh Seedless					
0	38.7	8.0	2.28	39.62	No pittings, good
1	32.3	9.0	1.88	41.76	— do —
2	39.0	7.0	1.81	39.33	— do —
3	40.0	8.0	1.35	38.74	— do —
4	38.8	8.5	1.19	40.00	— do —
Ruby					
0	46.8	9.0	1.50	41.73	No pittings, good
1	41.9	10.0	1.44	45.78	— do —
2	43.4	10.0	1.53	40.51	— do —
3	46.0	10.0	1.05	43.75	— do —
			2nd YEAR		
Saharanpur Special					
0	37.0	10.0	1.85	39.57	No pittings, good
1	36.7	9.0	1.65	51.36	— do —
3	38.3	10.0	1.47	44.18	— do —
5	39.1	9.0	1.22	46.80	— do —
7	26.7	8.8	0.90	51.90	No pittings, slightly soft and less juicy
Marsh Seedless					
0	39.4	7.0	2.14	42.95	No pittings, good
1	43.3	7.5	2.06	30.00	— do —
3	37.1	8.0	1.45	57.95	— do —
5	35.5	8.5	1.14	37.70	— do —
7	24.4	8.0	1.01	23.20	No pittings, slightly soft and less juicy
Ruby					
0	46.0	9.5	1.58	45.89	No pittings, good
1	44.1	10.0	1.33	56.61	— do —
3	38.1	8.0	1.26	48.33	— do —
5	41.6	8 0	1.04	46.48	— do —

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Shelf Life Extension of Fruits and Vegetables by Combined Use of Refrigeration and Gamma Irradiation

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The post-harvest deterioration in fresh fruits and vegetables, can be classified into three major categories: (i) biochemical and physiological changes of an endogenous nature; (ii) microbial and insect spoilage; and (iii) dehydration and mechanical injury. Although refrigeration materially retards the rate at which fruits and vegetables deteriorate from all these causes, with relatively little adverse effect upon taste, texture, nutritive value and overall quality, the fruits grown only in the temperate zones are more amenable to prolonged refrigerated storage. For tropical fruits like bananas and mangoes lowering the storage temperature, leads to cold injury and resultant loss of quality on ripening.

Extensive studies carried out at the Bhabha Atomic Research Centre have shown the potentialities of gamma irradiation either alone or in conjunction with other post-harvest procedures in enhancing the market life of certain fruits and vegetables[1-4].

The benefits accruing from irradiation processing was found to be maximal when it was combined with cool storage and therefore, could be a better alternative method to refrigeration, requiring lower temperatures. The present paper examines some aspects of cool storage and irradiation in extending the shelf life of certain ftuits and vegetables.

Studies on delayed ripening in Alphonso mangoes

The annual production of mangoes in India is about 3 million tons and the variety Alphonso, because of its distinctive qualities, is very popular with the canning trade as well as for consumption in the fresh state. Earlier studies at Trombay have shown that preclimacteric fruits given an optimum dose of 25 Krad could be kept in a marketable condition for 20 to 80 days when stored at temperature between 20° and 5°C[2, 5]. However, in recent studies, storage of Alphonso mangoes whether irradiated or unirradiated at temperatures below ambient was found to result in loss of the typical Alphonso flavour and decreased carotenoid formation on ripening.

Table 1 summarizes the organoleptic ratings of Alphonso mangoes held for varying periods at temperatures between 7° and 20°C and subsequently ripened at

TABLE 1. *Effect of post harvest temperature on organoleptic qualities of Alphonso mangoes (a).*

Storage and ripening conditions	Appearance	Aroma	Flavour	Texture	Average score
12 days at R. T. (27–32°C)	8.5	8.0	8.7	8.2	8.3
10 days at 20°C+5 days at R. T. (b)	6.1	5.7	6.4	6.0	6.0
28 days at 20°C	7.3	6.2	6.0	6.6	6.5
10 days at 15°C+5 days at R. T. (b)	7.7	7.0	6.8	6.8	7.0
28 days at 15°C+5 days at R. T. (b)	6.5	5.8	6.0	6.8	6.3
10 days at 7°C+5 days at R. T. (b)	7.4	7.0	6.3	7.0	6.9
28 days at 7°C+5 days at R. T. (b)	7.1	7.1	7.1	7.6	7.2

(a) Based on a 9-point hedonic scale, from 1 extreme dislike to 9, extreme liking. A score of 5.5 and above was considered acceptable.

(b) These fruits stored at the respective temperatures on removal were given a hot water dip (50° C for 5 min) and subsequently held at room temperature for ripening.

ambient temperature after subjecting them to hot water treatment. It can be seen that fruits stored and ripened at ambient temperature had better flavour and other quality attributes than those stored at lower temperatures. The loss in flavour and other quality parameters was noticeable even in fruits held for periods as low as one week at the respective low temperatures.

Apart from lowering the quality attributes, storage of Alphonso mangoes at refrigerated temperatures was also found to result in decreased carotenoid synthesis on ripening. Fruits stored and ripened at ambient temperatures had orange yellow flesh as compared to yellow flesh in those stored at low temperatures and subsequently ripened at ambient temperature. The total carotenoid content in fruits stored and ripened at ambient temperature was almost twice as that of fruits stored at lower temperatures (Table 2). These results suggest that though the preclimacteric life of Alphonso mangoes could be increased by refrigeration, the quality of the fruits was not typical of fruits held at ambient temperature throughout. Irradiated fruits held

TABLE 2. *Effect of post harvest temperature on total carotenoids content of Alphonso mangoes*

Storage and ripening conditions	Total carotenoids mg/100 g fresh pulp
12 days at R.T.	13.8 (12.5 — 15.8)*
28 days at 20°C	5.6 (4.5 — 7.5)
28 days at 15°C + 5 days at R.T.	5.8 (4.0 — 6.5)
28 days at 10°C + 5 days at R.T.	7.8 (7.5 — 9.6)
28 days at 5°C + 5 days at R.T.	8.4 (7.2 — 10.5)

Fruits stored at lower temperatures were given hot water dip for stimulating ripening. Values are average of 4 to 8 independent estimations.

*Figures in parenthesis show the range of values.

TABLE 3. *Effect of post harvest temperature on self-life of gamma irradiated bananas*

Variety	Maturity at harvest	Dose in Krad	Preclimacteric life in days: Control 13–14°C	Control 20°C	Control 27-31°C	Irradiated 20°C	Irradiated 27-31°C
Dwarf Cavendish	85%	30	16	10	5	16	7
Giant Cavendish	75%	35	19	11	8	17	11
	85%	35	14	10	5	15	7
	75%	25	14	9	4	16	7
Poovan							
	90%	25	10	6	3	10	5

under similar conditions also showed loss in quality and decreased carotenoid formation.

Shelf-life extension of bananas

The banana occupies an important place, second only to the mango, among the fruits cultivated in India, with annual production, of well over 2 million tons, most of which goes for internal consumption. Researches on the feasibility of employing gamma irradiation for shelf-life extension were carried out using several banana varieties such as Dwarf Cavendish, Giant Cavendish, Poovan, Red Skin and Rajeli (French plantain)[6]. Table 3 shows the results of irradiating bananas harvested at different stages of maturity. It can be seen that the shelf-life in the pre-climacteric state was maximum when the irradiated fruits were stored at 20°C and the storage life was comparable to that obtained when fruits were held at 13-14°C, the optimum storage temperature for these varieties.

Control of mold infection in harvested fruits

Unlike chemical fungicides, gamma irradiation due to its extreme penetration, can be used for treating deep seated pathogens within the host tissues to provide a therapeutic effect. However, the dose required for effective control of pathogens invariably results in undesirable changes in the host tissues. Recent studies at

Trombay have shown that a combination of irradiation and heat offers the most promising means of controlling fungal spoilage of fruits without adversely affecting the normal quality attributes and such combined treatments were more effective than either radiation or heat alone[1]. Fruits like bananas, mangoes, figs and grapes subjected to the combination treatments showed maximal beneficial effects when the storage temperature was lowered to 15°C (Table 4). Figs and grapes which are normally harvested at table ripe stage when subjected to the combination treatment showed shelf lives for periods upto 8 and 18 to 21 days respectively at 15°C, as against 4 and 10 to 14 days at ambient tempeiatures. Hot water dip, alone or in combination with chemical fungicides has been shown to be effective in delaying fungal incidence in mangoes[7-9]. However, heat treatment augments ripening, thereby reducing total shelf-life. Hot water dip combined with low doses of irradiation obviating the use of chemical fungicides thus seems a better process in that delay both in ripening and fungal infection can be achieved in climacteric type fruits like bananas and mangoes, the benefits being more at low temperatures. The organoleptic qualities of fruits receiving a combination of treatments were comparable to normally ripened fruits (Table 4).

Cool storage of irradiated potatoes

Out of the annual production of 4.5 million tons of potatoes, approximately 30 per cent (1.3 million tons) is estimated to be stored under refrigeration and the remaining crop is either held for short periods in village store houses or goes for immediate trade and distribution. Storage of potatoes at 2°–3.5°C prevents both

TABLE 4. *Extension of shelf-life of artificially inoculated fruits after treatment of optimum combination of gamma rays and hot water dip* (50° C, 5 min) (*a*)

Fruit	Treatment	Storage period (days) 27–32°C	Storage period (days) 15°C	Organoleptic score (b)
Figs	Nil	1	2	
	Heat + 150 Krad	4	8	6.75 (7.0)
Grapes : Anab-e-Shahi	Nil	2	8	
	Heat + 100 Krad	10	18	7.10 (7.30)
Grapes : Seedless	Nil	4	9	
	Heat + 100 Krad	14	22	7.70 (7.85)
Mangoes : Alphonso	Nil	10	10	
	Heat + 25 Krad	26	38	8.10 (8.50)
Mangoes : Langra	Nil	8	15	
	Heat + 25 Krad	16	26	7.50 (7.83)
Banana : Cavendish	Nil	7	13	
	Heat + 32 Krad	10	23	7.0 (7.33)
Banana : Mysore	Nil	4	9	
	Heat + 25 Krad	9	20	7.9 (8.5)

(a) Values are based on a 20 per cent wastage basis.

(b) Grapes and figs were served to panel members immediately after treatment, whereas mangoes and bananas were served when they reached Table ripe stage. Figures in parenthesis refer to fresh, ripe, untreated fruits.

TABLE 5. *Losses (per cent) in 'Kufri Chandramukhi' potatoes during storage at low temperature*

Days in storage	Unirradiated 15±1°C; 80–85% R.H. Microbial spoilage	Dehydration	Total	Irradiated 15±1°C; 80–85% R.H. Microbial spoilage	Dehydration	Total
30	0.1	2.3	2.4	0.2	3.0	3.2
50	0.6	5.1	5.7	0.9	5.0	5.9
80	0.8	8.9	9.7	1.0	—	—
120	1.0	14.3	15.3	1.2	8.2	9.4
160	1.1	22.3	23.4	2.0	9.5	11.5

Unirradiated tubers stored at 15°C commenced sprouting after 1 month and were in an unmarketable state at the end of 4 months. Values are based on 100 kg each of potatoes (5×20 kg) in each lot.

rotting and sprouting and the tubers could be kept in a marketable conditions for 6 to 8 months. However reducing sugars and sucrose accumulate in tubers in exessive quantities at temperatures 4°—10°C[10]. High reducing sugars in conjunction with nitrogenous compounds present in the tubers can cause undesirable dark brown colour of potato chips and french fries and the browning of dehydrated potatoes during storage. The lack of adequate cold storage capacity and the heavy cost involved also calls for alternative method of storage of potatoes at temperatures above 10°C.

Though chemical sprout inhibitors like maleic hydrazide (MH), methyl-ester of alpha naphthalene acetic acid (MENA) and isopropyl-N-chlorophenyl carbamate (CIPC) are being used in many countries, it is reported that these chemicals do not give satisfactory sprout inhibition when the storage temperature exceeds 25°C[11, 12]. Low dose (10 Krad) gamma irradiation offers a potential alternative method for complete prevention of sprouting in potatoes regardless of storage temperature. Apart from sprout inhibition, some of the other beneficial effects of gamma irradiation are: (i) decreased dehydration due to sprout inhibitions; (ii) reduced greening and solanin formation, occasionally encountered in cold stored potatoes exposed to light; and (iii) control of potato tuber moth.

One of the serious problems encountered in storage of potatoes under high ambient temperatures prevailing in tropical countries is the accelerated rate of microbial spoilage mainly due to *Erwinia carotovora*. Studies at Trombay have shown that microbial rot and sprouting could be successfully combated by storing irradiated tubers at 15°C. A perusal of the data in Table 5 would show that the losses due to sprouting, rotting and dehydration are significantly less in irradiated tubers held at 15°C as compared to unirradiated tubers. Thus irradiation followed by cool storage offers an alternative method for conventional cold storage (2–4°C) of potatoes.

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Liquid Nitrogen for Food Refrigeration

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Frozen Food Technology has undergone important changes in the 1960s, with the commercial availability and use of a range of cryogenic fluids, and the emergence of a new technology—Cryogenic Technology—dealing with the production and application of supercold temperatures, conventionally accepted as ranging below minus 150°C. The commercially important cryogenic refrigerents are: oxygen, nitrogen, argon, hydrogen and helium. Cryogenic machinery includes all plant and equipment for the production, storage, distribution and handling of cryogenic fluid.

Cryogenic technology is a by-product of the nuclear-missile-space age; it originated from crash programmes to support super projects in the fields of nuclear weapons and space missions. During the past 25 years, while weapons and space missions continue to be the focal points of interest, spectacular applications of cryogenic refrigeration to an ever expanding field of industries, have made cryogenics an important tool of present and future industries. These include nuclear power, electronics, satellite communication, transportation, metal smelting and processing, fertilizers, petro-chemicals, food preservation, navigation guidance, blood and biological preservation, etc. In the Indian context, several sophisticated industries have been set up during the past two decades, involving cryogenic processes, without a matched growth of cryogenic technology. Technological self-reliance in this field is an essential requisite to meet the confrontations of the space age.

A new standard of refrigeration capacity has been set by the use of cryogenic fluid refrigerents. Of these the most commonly used cryogenic fluid is liquid nitrogen: it has properties that make it specially suited for food preservation, freezing of foods etc. and for refrigeration for trucks, trailers, railcars for in-transit refrigeration. Liquid nitrogen is relatively inexpensive refrigerent, easy to obtain from air, chemically inert, will not burn, explode or leave a residue when it evaporates. With

new technological breakthrough in the science of cryogenics and the development of larger tonnage air separation plants, liquid nitrogen availability at commercial prices is continuously extending the use of liquid nitrogen refrigeration, at lower total cost than conventional mechanical refrigeration, especially for high-value foods.

Liquid nitrogen has a normal boiling point of —320.5°F, and a latent heat of 85.67 BTU per pound.

The significant features of liquid nitrogen that enables its use for food preservation are: (i) Its high refrigeration capacity, and (ii) precise control, as explained below:

(a) In the process of evaporation from liquid to gas at —320°F, every pound of liquid nitrogen, absorbs from the food	86 BTU
In warming up from —320°F to 0°F, (cargo temperature) absorbs	80 BTU
Total refrigeration capacity	166 BTU

This gives an enormous temperature driving force, thermodynamically instantaneous, enabling the initial cool-down period, from summer heat to sub-zero temperatures to be less than five minutes, whereas conventional mechanical refrigeration takes hours.

(b) The liquid nitrogen, when it becomes a gas, has a volume expansion of nearly 700 times. This facilitates thorough penetration and rapid heat transfer and precise temperature measurements and control of liquid delivery instrumentation for economic and efficient manual or automated operation.

The large temperature driving force and precise control to suit operating variables of the food material and its characteristics, places no limits to the applications of liquid nitrogen refrigeration to every variety of foods and perishables.

A good freeze preservation system will aim at retaining the characteristic appearance, aroma, texture, taste, vitamin content and all gustatory appeals, as near to the natural form as can be achieved.

Fresh food begins to deteriorate from the moment it is harvested, from bacteriological, enzymatic, oxidising and other reactions, which break down the structure of the food physically and chemically, adversely affecting flavour, nutritive value, shelf life, etc. The object of refrigeration is to stop the effect of bacterial action without breaking down the cell walls of the food material. Most foods start to freeze at about 30°F with ice crystals appearing. As freezing proceeds, the increasing concentration of dissolved constituents of the cell fluids, continuously lower the freezing point, and final freezing occurs at temperatures as low as -35°F. The slower the rate of freezing, the larger will be the ice crystals and their number in the intercellular space, leading to cell wall rupture and cellular fluid migration. Both during freezing and thawing some moisture drains away, carrying dissolved solids. A good freeze preservation system that preserves the quality of freshness economically without changing the cellular structure is the superfast freezing technique by liquid nitrogen which is a new field of applied cryogenics and commercially known as LN-IQF (liquid nitrogen-individual quick freeze).

The superfast LN-IQF technique for the food refrigeration industry, especially for the high value sea-foods like shrimps, has made profound impacts, and IQF product command premium prices in world markets.

The outstanding advantages of the LN cryogenic freeze system can be summed up as :

(*a*) In capital cost LN freezing system costs significantly less than the conventional blast or plate freezer system, because it involves no refrigeration equipment and associated horsepower, switch gear, and circulation piping. The LN supplier, instals and maintains the cryogenic equipment and supplies the nitrogen, usually on a monthly deferred system and 'pay as you earn' basis. As there are no moving parts, the maintenance costs as well as labour employed are considerably less.

(*b*) The plant occupies minimum space, has great mobility for location at site or on-board fishing trawlers, enabling rapid handling and distribution of products.

(*c*) Cryogenic freezing tunnels have least start-up and shut down time (less than 10 minutes) and allow maximum versatility of products and flexibility of product output.

(*d*) The weight loss from drip and dehydration associated with conventional blast freezing is an important consideration for food processors of high value foods. In cryogenic freezing such product weight loss is negligible. This reduction in product weight is often enough to compensate for the higher cost of nitrogen, besides improved texture, appearance and freshness quality.

On an overall cost analysis, taking into account product weight-loss from dehydration and drip loss, labour, operating expenses, overall quality, and versatility, liquid nitrogen refrigeration has definite advantages; the higher the value of the products processed, the higher the benefits.

The scope and prospects for liquid nitrogen refrigeration in the Fifth Plan : In the Indian context, liquid nitrogen application to food freezing has a significant role particularly in the field of seafoods export industry. A major part of export earnings from marine products arises out of the frozen shrimp industry. According to the Foreign Trade Ministry and the targets of the Fifth Plan, the export income from the frozen shrimp industry is expected to go up to Rs. 118 crores by 1979. Marine Products Exports Development Authority has been set up to undertake suitable measures for development and regulation of the Industry. The processing of shrimps in India is exclusively based on the conventional plate-type or blast freezing, which produces an inferior product. In the international market, India has to compete with several other countries, who employ modern cryogenic freezing and sophisticated equipment for shrimp processing. Such cryogenic freezing using IQF technique preserves, without damage the delicate cell tissues, freshness, flavour, vitamin content etc. beside reducing the processing time. As a result Indian frozen shrimp is heavily marked down in prices; in fact the Indian product is excluded from the premium priced quality class.

The deep involvement of cryogenic refrigeration and the growing sophistication and complexity of equipment in the frozen shrimp industry necessitates that a wide based support to develop indgienous technology and equipment, should be provided. In the absence of this the vast potentials of the seas surrounding India will remain a

monopoly field for overseas agencies who have developed the requisite technology and equipment.

A task force assigned to examine the problems and recommend measures for upgrading the status of the frozen shrimp industry in world markets, has provided guidelines for the modernisation and expansion of the shrimp industry. These guidelines include: (*i*) assistance to shrimp processors to change over to the LN-IQF technique, (*ii*) supply of the sophisticated technology and equipment, and (*iii*) supply of liquid nitrogen at subsidised rates.

Currently the NCST, NPL and several National Institutions as well as industry are interested in this field.

Freezing and Freeze Drying Characteristics of a Few Selected Foods

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Introduction

Soft drinks and fruit beverages are normally marketed as bottled products. In addition to the fact that about 80-90% of the total weight of a soft drink or a stabilised beverage is contributed by pure water, these preparations have a large bulk. Transportation of these consignments for consumption by military formations spread over large areas very often unconnected by well laid out roads poses a tremendous problem. The large bulk and weight and the vulnerability of glass bottles to transit breakage makes these products uneconomical from logistic considerations particularly when transport by air is involved. Fresh fruits also cannot be supplied under such situations because in addition to the weight consideration these are highly perishable and require a chain of refrigeration facilities from harvesting to the consumers end. Creation of cold storage facilities all along the supply line and at consumer centres for highly dispersed populations like Armed Forces is not only uneconomical but also not-practical.

The present communication reports attempts in the production of stabilized freeze dried fruit juice powders for possible use in certain military situations where bottled soft drinks cannot be supplied.

Materials and Methods

Materials preparation: Three different fruit products used in this study; were orange, pineapple and mango.

Freezing: Two trays of size 660×660 mm were kept in a blast freezer ($-20°C$). The fruit juice/pulp was poured into it at the rate of about 5 kg per tray to a

thickness of 12 mm. A few copper constantan thermocouples were placed in the middle of the product, at a height of 6 mm from the bottom, and connected to a potentiometer calibrated to measure temperatures. The other ends of the thermocouples were placed in melting ice (0°C). The time temperature study for orange juice is shown in Fig. 1. It took about 120-240 min for the fruit pulp to solidify sufficiently for cutting into blocks for further processing (4°C). The frozen material was cut into bits of 25×25×12 mm size inside a walk in cooler (4°C). These bits were then arranged in three drying trays (600×66) mm) to a loading density of about 8.0 kg/m². A few thermocouples were again attached to the surface and a few embedded into the geometric centre of the slabs and frozen at —30°C.

Freeze drying : Pilot plant model freeze dryer of "Socaltra" make was used in these studies. There was provision to monitor a weight loss from one of the trays continuously over the entire drying period. Temperatures of the material both on the surface and deep inside and also of the heating medium and condensor plate were recorded every two minutes through an Otic Fisher Porter recorder using copper constantan thermocouples. Drying was assumed to be complete when the deep temperature and the surface temperature met at 55°C and maintained at that temperature for atleast 30 min. This was further confirmed when there was no weight loss from the tray that was weighed continuously for half an hour. The vacuum was broken with a dry inert gas and the material was scraped from the trays and packed immediately in cans under nitrogen.

Compression : Since freeze dried material had a low bulk density an attempt was made to increase it by compression, using a Carvar press. 100 g of dried material were taken in a mould of 40×40 mm cross section and compressed under different loads of pressure. Initial and final volume of the material, the physical stability of the block prepared and also time taken by the block to dissolve completely when left undisturbed in 500 ml of water, are shown in Table 1.

Results and Discussion

The juices were prepared at different concentrations by adding cane sugar and

TABLE 1. *Compression studies on orange juice powder*

Weight = 100 g ; Volume = 492 cm³ ; Area = 4 × 4 cm

Pressure psi	Length of the block cm	Compressed volume cm³	% of original volume	Time (min.) to dissolve in 500 ml of water	Stability
1000	10.5	128	33.5	0.5	Too brittle
,,	9.5	152	31.0	0.5	,,
1500	9.5	152	31.0	1.0	,,
,,	8.5	136	27.5	2.0	,,
2000	8.5	136	27.5	3.5	Brittle
,,	8.5	136	27.5	4.0	,,
2500	7.5	120	24.4	4.5	Not brittle
,,	7.8	125	25.5	4.5	,,
3000	7.0	112	22.7	5.0	Hard
,,	7.0	112	22.7	5.0	,,

their freezing and freeze drying characteristics studied. As expected there was a depression in the freezing point[2] corresponding to the increase in the total soluble solids (Fig. 1). The time taken for cooling the product to a maximum temperature of complete solidification increased as its concentration increased; it was 120 min for 15° Brix, 150 min for 25° Brix and 18) min for 35° Brix in the case of orange juice. At this temperature the viscosity of the product rises to an extent where it is apparently solidified though not completely frozen. According to Louis Ray[3] this maximum temperature of complete solidification is both a necessary and sufficient factor. According to Riedel[8], at —30°C, 96% of the total water present and 100% of freezable water in any fruit or vegetable juice, is frozen. In Fig. 1 it is observed that the thermal arrest time was sharp and long in the case of lower brix than with higher brix. This is probably due to the ease with which water crystals could be formed in juices of lower brix because of latter's physical properties.

In freeze drying, juices of higher brix presented a different problem. Because of their lower cryohydric point the product should be frozen to a lowest possible temperature before heating could be started. The rate of application of heat to sublime the moisture also had to be lower throughout the period of drying. With increased rate of heating insipient melting occurs[4]. Because of low pressure in the

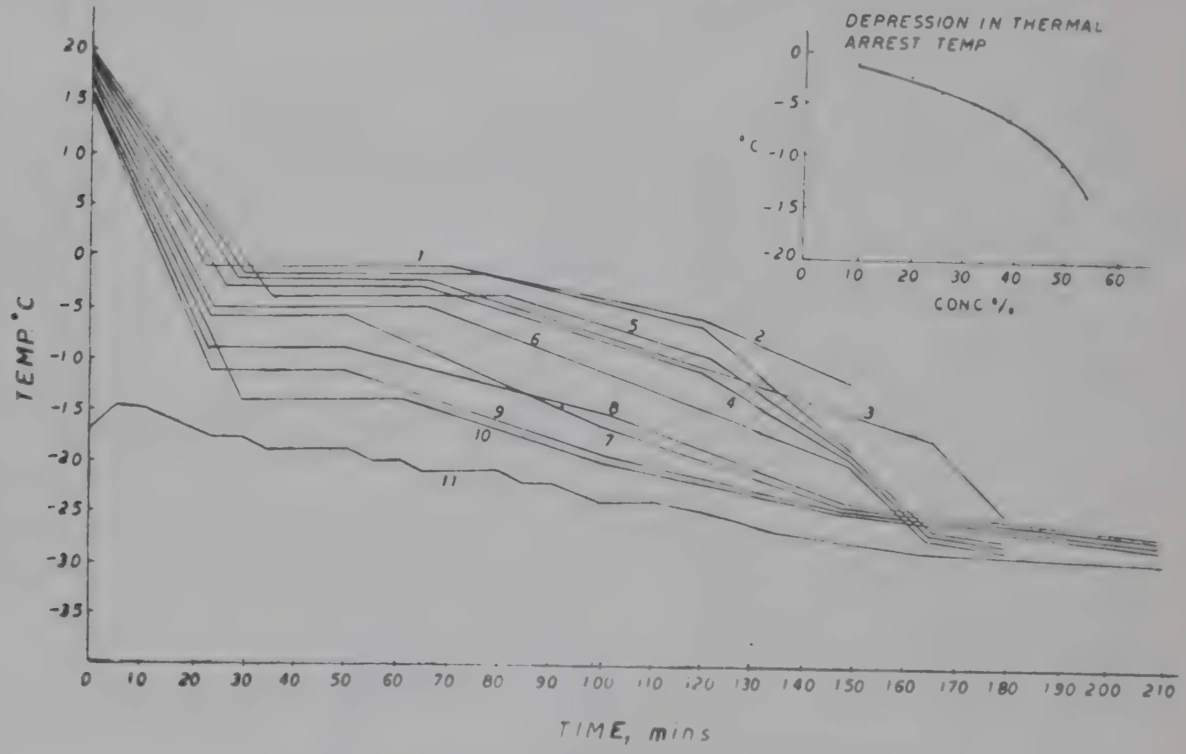

Fig. 1. Freezing characteristics of orange juice

Brix °	Brix °
1 — 11	7 — 40
2 — 15	8 — 45
3 — 20	9 — 50
4 — 25	10 — 55
5 — 30	11 — Condensor Temp. °C
6 — 35	

TABLE 2. *Freeze drying characteristics of few selected fruit Juices*

Items	Orange				Mango	Pineapple
	15°	20°	25°	19°	19°	20°
Wet wt. (kg)	9.3	10.6	10.6	10.1	8	10.9
Dry wt. (kg)	1.6	2.4	2.7	2.0	1.7	2.2
Drying time (hr)	11.0	10.0	8.0	11.0	8.5	8.0
Loading density (kg/m²)	8.0	8.2	8.0	7.0	6.9	8.2
Initial moisture (%)	83.0	78.0	68.0	79.0	78.0	78.0
Final moisture (%)	2.0	2.0	2.0	3.0	2.0	2.0
Max. surface temp. (°C)	45.0	45.0	45.0	45.0	50.0	48.0
Thickness (mm)	12.0	12.0	12.0	12.0	12.0	12.0
Abs. pressure (mm of Hg)	0.3	0.3	0.3	0.3	0.3	0.3
Peak sublimatic rate	1.4	1.4	1.3	1.4	1.5	1.6

chamber unregulated puffing starts and hinders further heat and mass transfer. Moy[5] after his experiments concludes that vacuum puff freeze drying could give good quality juice powder provided there is proper control of both initial temperature and the degree of vacuum. The present authors have observed that though controlled puffing helps drying rate, bigger bubbles tend to increase the vapour pressure inside and cause liquid phase drying.

Table 2 gives the freeze drying characteristics of fruit juices under study. For the same loading density, the total time taken for drying is less in the case of higher concentrated juice because of less amount of water to be sublimed. This is in spite of the fact that the rate of application of heat could not be as severe as in the case of juices of lower concentration. This is contrary to the finding of Foda *et al*[1] who have reported that the drying time increases as the concentration of the juice is elevated, due to the physical structure of the concentrated samples. Though physical structure may contribute in retarding the drying rate, total quantity of water to be removed is far less in juices of higher brix and this should bring down the drying time considerably as recorded in the present studies. Another reason for having a lesser drying time in the case of juices of higher brix is that in a system of freeze drying with programmed platen temperature (Fig. 2), approximately three fourth of water is sublimed during the first four hours. The remaining one fourth of water is sublimed during the falling rate period. Thus more quantity of water is to be removed during the falling rate period in the case of juices of lower brix and hence longer time for drying.

A concentration of 20° Brix was found to be optimum for preparing ready-to-serve beverages. Ole Moller[6] has suggested a concentration of 20°-30° Brix for industrial freeze drying of juice. Ease and economics of drying and also a good dilution ratio confirms the choice. There does not seem to be much difference in the peak sublimation rate values of different fruit juices.

Efforts to increase the bulk density of freeze dried juice powders by compressing them into blocks yielded encouraging results. Rehman *et al*[7] have attempted different compression ratios from 4 : 1 to 16 : 1 for vegetables like peas and

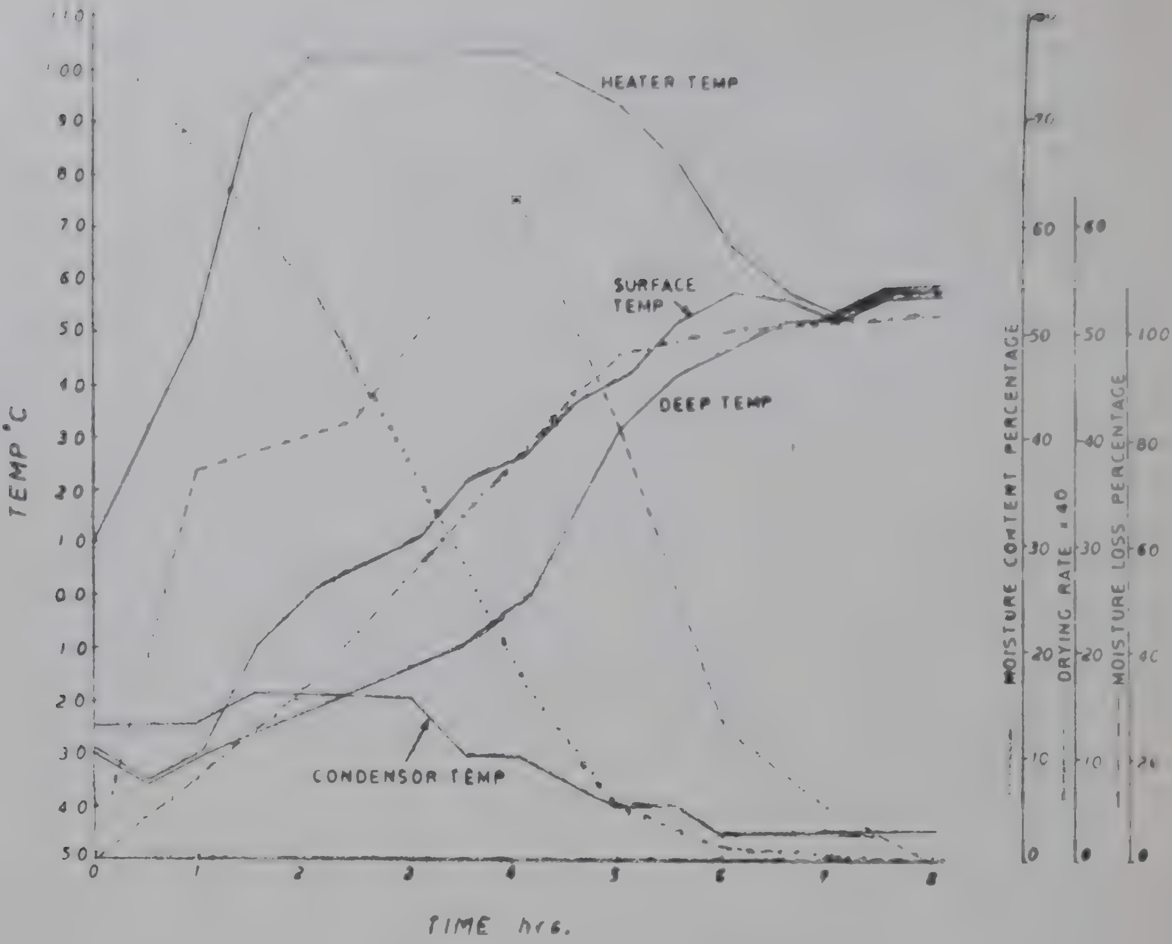

FIG. 2. Freeze drying characteristics of pineapple juice — 20° Brix

carrots. In the present studies reversible compression is used in the case of fruit juice powders because of its ultimate use as ready-to-serve beverage. Table 2 gives the results of such experiments. A pressure of 2500 psi, brought the volume of an otherwise bulky and porous freeze dried powder to one fourth of its original volume. No conditioning of moisture content was attempted before compression. Blocks subjected to such pressure dissolved easily in about 4½ min when left undisturbed in 500 ml of water at room temperature (25°C). These blocks are easy to handle and to pack in flexible pouches.

Tables 3a and 3b give a comparison of advantages of freeze dried ready-to-serve beverages with other commercially available beverages. Though the cost per kg of

TABLE 3 (a). *Comparison of commercially available fruit juice beverage with freeze dried ready-to-serve beverage based on fruit juice*

Basis : 200 ml or ready-to-serve beverage

	Crush	Squash	R.S.B.	Freeze dried R.S.B.
Gross wt., g	93	124	570	28
Cost, Rs.	0.62	0.62	0.80	1.40

TABLE 3 (b) *Details of commercially available fruit juice beverage*

Details	Crush	Squash	R.S.B.
Quantity (ml.)	650	650	200
Conc.° Brix	55	45	15
Fruit juice (%)	25	25	5/Nil
Dilution	1 : 4	1 : 3	Nil
Acidity	1.4	1.3	0.25
Gross wt., (g.)	1200	1200	570
Cost, Rs.	8.00	6.00	0.80

solids is high in case of freeze dried powders, other logistic advantages are considerable. Weight and volume per kg of solids in case of freeze dried powders is very low and the problem of breakage of glass containers is not there. Freeze dried ready-to-serve beverage, based on fruit juices, have therefore, an edge over the commercially available beverages when supplies to remote areas are contemplated.

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Utilization of Natural Climatic Conditions for Freezing of Perishable Products in High Altitude Mountainous Areas

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Supply of fresh fruits, vegetables and meat to troops deployed in high altitude mountain areas presents considerable difficulty as they are snow bound for the major part of the year, and often completely cut off from the base supply points during winter months. Though the climatic conditions during winter months are unfavourable for the growth of microorganisms responsible for spoilage, large diurnal variations in ambient temperature (+10 to −25°C) cause cyclic freezing and thawing which result in the destruction of natural cellular systems of the food items. Under natural conditions, the slow rate of freezing leads to the formation of large ice crystals

which result in alteration of the ultrastructure of the tissue, denaturation of protein, drip losses etc. The changes in physical, chemical and sensory properties render the food unacceptable to the consumers.

Foods stored under fluctuating temperatures undergo rapid deterioration as some enzymic and nonenzymic reaction rates increase considerably.[1] Hence products stored under natural climatic conditions do not keep satisfactorily.

Fresh fruits, vegetables and meat after suitable treatments and packaging can be preserved for sufficiently long periods under freezing conditions (preferably below -18°C)[2]. In Great Lakes region of United States making use of the natural snow for maintaining uniformly low temperature, fish is preserved in a frozen state. This method of preservation is known as 'weather freezing' and the 'weather frozen fish' are reported to command good market. In such areas, where the natural snow is not available in sufficient quantities though the temperatures as low as -25°C may be common during winter months, fruits and vegetables are stored with thermal insulation in bunkers. Temperature range of 0—5°C is obtained by utilizing the heat of respiration of stored fruits and vegetables. Field storage trials of meat, vegetables and fruits carried out during winter months utilizing these two methods are presented in this paper.

Freezing Preservation of Meat by Burying in Snow

Carcass buried deep in snow was unsatisfactory as the product developed undesirable odours within a short period of storage. Preservation of perishable products under natural climatic conditions by maintaining lower temperatures in specially designed container using the natural snow mixed with salt was investigated. A double walled aluminium container (Fig. 1) filled with snow-salt mixture (78 : 22)

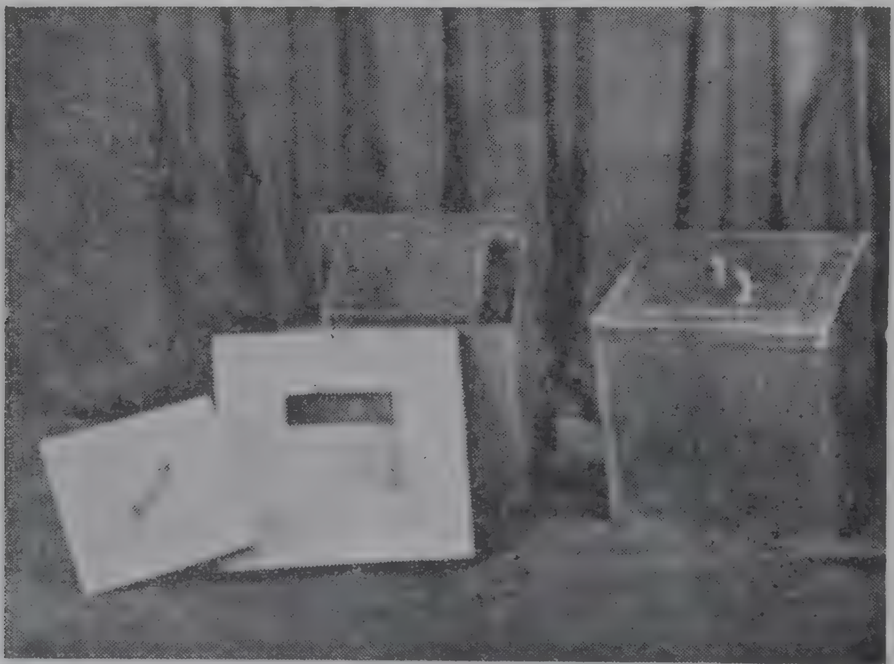

FIG. 1. *Double walled aluminium container*

TABLE 1. *Metereological data*

Months	Ambient temperature (°C)					Temperature (°C) inside the storage container			
	Max.	Average max.	Min.	Average Min.	RH %	Max.	Min.	Average	RH %
January	7.5	0.9	−16.5	−8.2	50–84	−15 5	−25	−21 9	90–95
Febraury	4.0	0.2	−17	−8.7	60–87	−15	−24	−19.8	,,
March	9.5	4.7	−13	−4.3	54–83	−15	−21	−18.9	,,
April	14	7.8	−3.0	+1.8	48–85	−16	−21	−16.4	,,

in the annular space was used. The perishable foods were filled into this container and burried 3 ft deep in snow. Though the ambient temperature ranged from +12 to —20°C, the temperature inside the container was near about —15°C on most of the days except when fresh snow-salt mixtures were charged at which time the temperature decreased considerably (Table 1). Since it is possible to preserve perishable products by this method of storage, further trials were carried out under actual field conditions (9000 – 12000 ft) using meat, fruits and vegetables. The details are given below :

Meat : Sheep, goat and chicken meat were tried. The animals and birds were starved for 24 hours prior to slaughter. The dressed carcasses were stored at —5° to —10°C overnight, and cut into small pieces (3" size). Meat with and without bones and fat were included in the study. A few samples of dressed meat were given a coating of distilled acetylated monoglyceride (Myvacet) by hot-dipping method. One sheep was given an antimortem injection of oxytetracyclin (10 ppm) three hours before slaughter to test the efficacy of the antibiotic treatment under these conditions.

The prepared meat (1 kg) was packed in different types of flexible packages (Table 2) and heat sealed. The flexible packages containing the meat were repacked inside 3-ply corrugated board cartons (11"×11"×11"), waxed externally, and placed inside the double walled containers containing the snow-salt mixture in the annular space.

The packed containers were buried three feet deep in snow at three places

TABLE 2. *Shelf-life (months) of different meats*

Meats	Packing			
	Cellophane polythene	Paper foil polythene	Myvacet coating and packed in poly	Sorbic acid treated wrapper and poly
Sheep dress	3½	3½	3½	4½
Sheep dressed deboned, defatted	4	3½	3½	5
Sheep injected with antibiotic and deboned and defatted	5	—	—	—
Goat dressed	3½	3½	3½	4
Chicken dressed	3½	3½	3½	4

situated at altitudes of 9000 ft, 11000 ft and 12000 ft above mean sea level. The maximum and minimum ambient temperatures at the trial centres and inside storage containers were recorded daily. Afterwards once in every 15 days, samples were analysed for microbial count, moisture content, peroxide value, free fatty acids and acceptability by the troops.

The periods for which different meats remained in good condition on the basis of organoleptic, physico-chemical and microbiological tests are given in Table 2. Sheep, goat and chicken meat can be preserved for more than 3½ months. Pre-wrapping of meat in wrappers treated with sorbic acid or an antimortem injection of oxytetracyclin (10 ppm) help in further extending the shelf life up to 5 months. The moisture loss in the meat during a storage period of 5 months was about 4%. During storage, the stored meat samples tended to become slightly softer in texture especially after 3½ months but there was no appreciable difference between these and fresh meat after cooking. Very slight off odour was detected in poultry meat after 3 months. The peroxide level was much higher in poultry meat than in sheep or goat meat during storage (Fig. 2). This could be expected as the poultry meat contains much higher level of polyunsaturated fatty acids. Consequently the development of oxidative rancidity in frozen poultry muscle would occur much faster than goat and sheep muscles. Peroxide value remained low upto three months storage but increased significantly thereafter, registered a maximum, and subsequently declined to a minimum in all three types of meat studied. The same type of relationship between peroxide value and storage period has been observed previously[1].

The free fatty acid content in the fresh meat was 0.5% in sheep meat, 0.6% in goat meat and 1.0% in chicken meat and varied from 0.6 to 1.5% during storage. Previous workers have reported that F.F.A. level in frozen meat increases when the

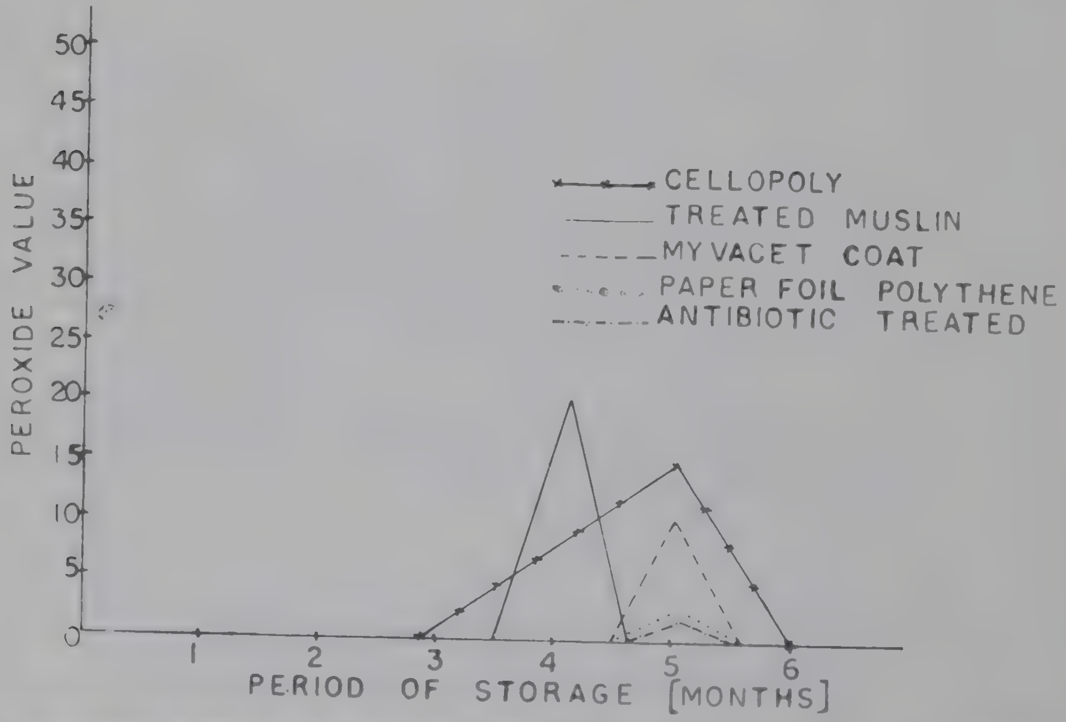

FIG. 2. Relationship between peroxide value and storage period of sheep meat (without bones/fat)

Table 3. *Particulars of fruits and vegetables used*

Fruits/Vegetables	Variety/Quality	Age when procured
Apples	Amri variety grown in Shukian gardens, Srinagar	2 months
	Russet variety grown in Hazratbal gardens, Srinagar	4 months
Oranges	Nagpur variety	2 months
Mosambie (Malta)	Unknown source	Not known
Cabbage	Grown in Dal Lake gardens, Srinagar	2 days
Carrots	Grown in Batmel gardens, Srinagar	2 days
Cauliflower	Snow white type with little browning. Grown in Jammu gardens.	3 days
Potatoes	New crop, small variety outer skin slightly reddish	2 months
Raddish	Grown in Gargihal gardens, Srinagar	2 days

temperature of storage is above —18°C but remains practically constant below this temperature.

Fruits and vegetables : Particulars of fruits and vegetables used are given in Table 3. The vegetables and apples were cleaned, blanched, sliced and packed in polyethylene bags (20 cm × 20 cm) which in turn were repacked in card board cartons, and waxed. Citrus fruits were peeled, juice extracted, filtered through muslin cloth, and fortified with ascorbic acid (1%) and sugar (4%). The juices were filled in high density polyethylene bottles and pasteurised at 70°C for 15 minutes. Waxed cartons and high density polyethylene bottles were packed into the aluminium double walled container and stored as described under meat. At intervals of one month the samples were tested for microbial count, organoleptic quality, moisture losses and ascorbic acid content.

The total microbial count in the vegetables was very high at the time of procurement but reduced considerably during washing and blanching. During frozen storage, further reduction was observed and at the end of the time at which their frozen product was satisfactory, the count was less than 10,000 per g. The storage life based on microbiological and organoleptic tests is given in Table 4.

Moisture content remained practically constant during storage. The losses in ascorbic acid in vegetables during washing and blanching varied from 6 to 72% and during storage from 72 to 90%. In citrus juices ascorbic acid losses during pasteurisation was 20% which progressively increased to 50% during frozen storage for 5 months.

The determinative factor in the shelf life of frozen fruits and vegetables in the present study was the acceptability of the product by users. Cabbage, carrot and cauliflowers remained practically unaffected during the entire period of storage (Table 4). Radish was acceptable upto 4½ months of storage but became slightly leathery in texture after 2 months, and required slightly higher cooking time. Carrots tended to become softer in texture after 4½ months but was acceptable. Amri and Russet varieties of apples remained acceptable upto 4 months but tended to become brown after 2½ months and their acceptability also decreased. Fruit juices with

TABLE 4. *Pretreatment and storage life of fruits/vegetables preserved by freezing*

Fruits/vegetables	Blanching time min	Storage life (months)
Apple (*a*) Amri variety	2	3.5
(*b*) Russet variety	2	3.5
Orange juice	—	4.5
Mosambic (Malta) juice	—	5.5
Cabbage	4	5.5
Carrots	6	5
Cauliflower	6	4.5
Raddish	10	4

added sugar and ascorbic acid remained in good condition during the entire period of storage (5 months) but the taste was slightly bitter without sugar.

Clamp Storage of Fruits and Vegetables

Fresh fruits and vegetables remain alive even after harvesting and respire during storage. At ordinary temperature, the respiration rate is high, and can be preserved only for a short period. However, if the temperature of storage is lowered (0-5°C), the respiration rate is decreased and their shelf life is extended. At temperatures below the freezing point, irreversible changes take place. The products exposed to temperatures near the freezing point can no longer respire. Therefore, to keep in the original state, the fruits and vegetables have to be stored above their freezing points. Although optimum temperature for cold storage of various fruits and vegetables vary, most temperate fruits and vegetables can be preserved for about 3-6 months at 0-5°C. Since the ambient temperatures during winter months in forward mountain areas in India vary from 10° to -25°C, fruits and vegetables have to be protected from freezing. In the present study by storing fruits and vegetables in specially constructed underground cellars or by burying in pits having a layer of wheat straw on all sides the heat of respiration of fruits and vegetables was trapped and freezing prevented. In underground cellars, the fruits and vegetables were stored in wooden boxes, polyethylene pouches or merely spread on the floor. Methods of storage in pits and underground cellars, and the shelf life for various fruits and vegetables are given in Table 5. The main problem in this technique is the selection of the exact thickness of insulation layer that has to be provided so that a uniform temperature of 0-5°C is achieved. If the insulation layer is too thick, the temperature inside the storage chambers will progressively increase leading to excessive respiration and spoilage and if too thin, spoilage occurs due to freezing injury. By this technique, apples, turnips, knol-khol and carrots could be preserved for 2 months, potatoes and radish upto 6 months, tomatoes for 1 month, and onions for 4 months.

In conclusion, it may be said that by the storage procedures described, it is possible to preserve perishable products in areas where ambient temperatures as low as -25°C prevail for about 4-6 months in the year.

TABLE 5. *Storage data of fruits and vegetables preserved by clamp storage method*

Vegetable/fruit	Method	Storage life days
Apple	Wrapped in paper and stored in wooden boxes.	30
Tomatoes	(1) In polythene pouches with or without ventilation	15
	(2) In wooden boxes with or without straw	,,
Cauliflower	Control	30
	In ventilated and unventilated polythene pouches	60
	Hung from ceiling	120
	Transplanted	120
Cabbage		120
Onion	Spread on the ground	120
	Hung from ceiling	120
Turnips	In pits with straw padding both at bottom and on top (10–15 cm thick) and covered by earth upto one foot in thickness	30
Carrots	In pits with straw padding both at bottom and on top (10–15 cm thick) and covered by earth upto one foot in thickness.	30
Raddish	(*i*) In pits with straw padding both at bottom and on top (10–15 cm thick) and covered by earth upto one foot in thickness.	120
	(*ii*) In pits in layers with 15 cm of earth in between two layers and the pit itself being lined with straw at the bottom and top.	180
Potatoes	In pits as in the case of turnips	180
Knol-khol	In pits as in the case of turnips	60

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Refrigerated Storage of Mandarin Oranges

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The area under citrus in India is estimated at 105, 396 hectares[1], and nearly 11,100 hectares are under Mandarin oranges in the State of Karnataka. The

Mandarin oranges grown in Coorg area alone are estimated at 155,000 tonnes per year[2]. There are two crops in this area, the commercially important one being harvested during November to April forming 60% of the total crop. The glut during February—March causes a fall in the price of the commodity and results in loss to the growers and merchants. The Mandarin oranges are highly perishable if stored without proper treatment and under less favourable conditions. They also tend to lose their texture and quality at ambient storage temperature and become unmarketable during prolonged storage. It was reported that waxing of mandarin oranges[3,4] was beneficial in extending their economic storage life by three weeks at ambient temperature enabling export to Singapore by sea without refrigeration. However, prolonged storage for over two months needs refrigeration. Optimum cold storage condition was reported to be 45°F, R. H. 85-90 per cent for Coorg Mandarin oranges[5]. The authors[5] have also observed wastage of about 10 per cent after 60 days of storage at 45°F. The fruits harvested in the months of March and April (when the temperature is 28-34°C) tend to shrivel faster, become unmarketable and fetch low prices. If these fruits are stored for more than two months in as near the fresh condition as possible, they fetch better returns duting the off-season. Therefore, studies were undertaken to find out the effects of low temperature on the Mandarin oranges with diameter (equatorial) varying between 2 to 3 inches (5-7.5cm.) during 75 days of storage.

Materials and Methods

Mandarin oranges grown at Pollibetta in Coorg district were harvested in the month of March. They were free from any apparent injury, bruises, blemishes, insect punctures, oil cell breakdown or any other disorder. Fruits were held in a perforated basket (150 number at a time) and dipped for 60 seconds in a tank containing 20 litres of wax emulsion (4.8 per cent wax solids) and 0.2 per cent Tecto-60, a fungicide. The fruits were removed, drained and dried in open air by spreading evenly on the floor over palm mats and by frequent turning. The dry, waxed fruits were packed in cartons with half the number of fruits wrapped in 15cm × 15cm tissue paper. Each corrugated cardboard carton (47 cm long, 28 cm wide and 28 cm. high) was provided with four holes (1.25 cm. in diameter) along the length on each side. A groove 3.5 cm. long and 1.25 cm. wide was provided on each broad end of the box one inch below the top to facilitate lifting of the carton. In addition to the groove one 1.25 cm. diameter hole was provided below the groove 2.5 cm. from the bottom. One hundred Mandarin oranges were packed in four layers in each carton with card board separators (46 cm. × 26.6 cm.) between layers. The flaps of the cardboard carton at the top were closed and sealed with gum tape. The cartons were weighed individually before and after packing. Two thousand fruits packed into twenty cartons were then loaded into a pick-up van in a length wise fashion and transported to Mysore by road, a distance of 55 miles and stored at 45°F, R. H. 85-90 per cent within 36 hours of harvest. Their storage behaviour in respect of losses in weight, spoilage, shrinkage, per cent juice content, total soluble solids, acidity, sugars and vitamin C was noted at the end of the storage period of 75 days. Their marketability was judged by firm texture, gloss and quality. Per cent

juice content in stored Mandarins was determined by AOAC method[6] and for any determination 20 fruits from each carton consisting of five fruits selected at random from each layer were used. Total soluble solids were measured by hand refractometer. Acidity was estimated as citric acid by the AOAC method[6]. Moisture in peel was estimated by drying 10 g. of the chopped peel at 60°C to constant weight. Sugars were estimated by modified Somogyi's method[7]. Observations on the desiccated segments were made on randomly selected fruits from each carton. The equatorial diameter of each fruit was determined before peeling. Desiccated segments were recorded for groups having diameter above 5 cm. but below 6.25 cm. and between 6.25 and 7.5 cm.

Results and Discussion

The results of trials carried out in the main season of 1973 are given in Tables 1, 2 and 3. From Table 1 it is evident that losses in weight in tissue wrapped fruits is slightly lower than those without tissue wrapping. The weight losses were as high as 10 per cent at the end of the storage period of 75 days; the spoilage was around 2 per cent. The marketable fruits at the end of the storage of 75 days were over 92 per cent and the unmarketable ones slightly over 5 per cent. The fruits had come down with *Penicillium* rot.

Table 2 shows that there was little change in the composition of the fruits as seen in peel moisture, acidity and vitamin C. In Table 3 are listed the number of segments per fruit, the number of intact segments and desiccated segments. The desiccation could not be seen in the whole fruit. It was noted on peeling that fruits with diameter less than 6.25 cm. had nearly 20 per cent desiccated segments whereas those with 6.25 to 7.5 cm. diameter had only 8.5 per cent desiccated segments. Similar results were found both in unwaxed and waxed fruits. It is interesting to note here that for prolonged storage over two months at low temperature, fruits with 6.25 to 7.3 cm. diameter are more suitable. The desiccation in the segments of the fruits might be attributed to the volume of segments and the number of juice cells present. Smaller the volume of the segment and greater the number of juice cells, the chances for segments to collapse and desiccate are high due to physiological stress. The post storage life of Coorg Mandarins stored for 75 days was also studied at ambient temperature. Based on the appearance, shrinkage, quality and texture,

TABLE 1. *Per cent weight loss, spoilage, marketable and unmarketable fruits seventyfive days after storage*

No. of cartons	No. of fruits	Additional treatment	Wt. loss (%)	Spoilage (%)	Marketable (%)	Unmarketable (%)	Total UM %
10	1000	Fruits wrapped	9.42	0.9	94.4	4.7	5.6
10	1000	Not wrapped	10.27	2.2	92.0	5.8	8.0
Difference with result				1.3*		1.1 n.s.	2.4*

*Average percent of fruits for which discount has to be allowed is 7; UM—Unmarketable. n.s.—Not significant.

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TABLE 2. *Chemical changes in Mandarin oranges*

Storage period (days)	T. S. S. (°Brix)	Juice content (%)	Peel moisture (%)	Acidity ascitric Acid (%)	Vitamin C (mg/100 ml)	Total sugars (%)
Initial @	15	43.9	68.6	0.60	38.5	14
Final ‡	15	40.5	64.1*	0.74	36.0	14

‡Estimated from four hundred oranges by drawing at random 20 fruits from each carton, five from each layer ; *No significant difference between wrapped and unwrapped fruits ; @Estimated from one hundred oranges.

TABLE 3. *Size of the Mandarin oranges, total number of segments per fruit and number of segments intact and desiccated after 75 days of storage*

Diameter of the fruit (D)	Total no. of segments/fruit	Total number of segments examined	Total number of segments	
			Intact	Desiccated
5 cm. but less than 6.25 cm.	10	240	192	48 (20%)
6.25 to 7.5 cm.	10	1260	1154	106 (8.4%)
Difference with result				10.8

the cold stored oranges were found to remain in good acceptable condition for a period of five days, after which deterioration set in rapidly.

Economics of Treatment

Cost of fruit : includes harvesting, grading, labour and tissue paper charges	Rs. 15.00/100 fruits
Cost of waxing and drying	Re. 00.50/100 "
Cost of each carton	Rs. 6.00
Cost of transportation of fruits from Pollibetta to Mysore	Re. 1.00/carton of 100 fruits
Cold storage charges for 2½ months at the rate of Rs. 1.20 per carton per month	Rs. 3.00
Miscellaneous expenses	Re. 0.50
Total cost	Rs. 26.00/100 fruits
Selling price during June 1973	Rs. 30.00
Discount on account of unmarketable fruits	Rs. 2.10 for 7 fruits
Cost of carton	Rs. 6.00
Total selling price with carton	Rs. 33.90
Total expenses incurred	Rs. 26.00
Amount recovered after selling	Rs. 33.90
Profit	Rs. 7.90/cartron of 100 fruits

Distributor has a good margin on this as seen from the above cost estimate.

Summary

Coorg Mandarin oranges harvested in March and treated with wax emulsion prior to packing in cartons, and stored within 36 hours after harvest at 45°F, and R. H. 85-90 per cent for a period of 75 days had over 93 per cent marketable fruits. Mandarin oranges of 5 to 6.25 cm. diameter had 20 per cent desiccated segments as against only 8.3 per cent desiccated fruits with equatorial diameter between 6.25 and

7.25 cm. Storing oranges for 75 days at low temperature was found to be profitable as the fruits fetched better price during the off season. The fruits were acceptable in gloss, appearance, texture, colour and quality. Their post storage life at ambient temperature after removal from cold storage was found to be five days.

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Studies on Low Temperature Breakdown in Mangoes

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Low temperature storage is universally acclaimed as the only practical and economical method of extending storage life for perishable produce. Several reports have appeared in literature suggesting optimum low temperature conditions of storage for fruits and vegetables[2,3,6]. But, many fruits and vegetables of tropical origin are susceptible to low temperature injury, collectively termed as 'Chilling Injury' or 'Low Temperature Breakdown'. This subject is comparatively little understood and most of the available literature is descriptive.

Low temperature breakdown (LTB) is entirely distinct from freezing injury as it appears in tissues exposed to a temperature above the freezing point, but lower than the ambient conditions. It manifests as skin blemishes, necrotic lesions, discolouration of the vascular bundles, and failure to ripen normally after removal to ripening temperature resulting in loss of colour, odour, flavour and taste. These types of injury are entirely physiological, although fungal attack may occur, following the primary injury.

Storage and transport of mangoes under refrigerated conditions, on a commercial scale as developed for the banana industry have not been successful hitherto, since this tropical fruit does not develop colour, odour and taste characteristic of a cultivar, although it is free from visible symptoms of external chilling injury[7,10].

Hence, intensive investigations were made on *Alphonso*, *Pairi* and *Neelum* cultivars and several pre-and post storage treatments were examined to minimise low temperature breakdown in mangoes. The restults on *Alphonso* cultivar are reported here

Experimental

Several experiments were conducted during the seasons of 1966 70. Green, firm and mature fruits were harvested from nearby orchards for these trials and were used within one day of harvest for different treatments. Fruits were stored in ventilated wooden boxes (50 per box) at appropriate temperatures for storage or ripening (85-90% R.H. at 35-55°F and 45-65% R.H. at 70-80°F). Chemical analysis and physical observations were made at regular intervals until the fruits were unmarketable and consumer acceptability trials were conducted when the fruits were soft and edible ripe. Promising treatments were repeated on a pilot scale on one tonne of fruits. For the purpose of brevity, results obtained on *Alphonso* cultivar are discussed, since the trend of results were similar in *Pairi* and *Neelum* cultivars.

Results

Studies on storage of mangoes at different temperatures have revealed that *Alphonso* cultivar develops symptoms of chilling injury in the form of sunken spots and as brown lesions on the surface of the fruit at temperatures below 55°F. The severity of this physical damage becomes more prominent with increased storage period and reduced temperature and the lesions coalesce to form larger brown patches. The fruits, however, remain green and firm at temperatures below 55°F for 22 days or more without physical damage and do not ripen satisfactorily when these are transferred to ripening temperatures (70°-80°F). They are easily susceptible to fungal diseases. The fruits become soft, but lack odour and flavour, characteristic of the cultivar and are unmarketable. Fruits stored at temperatures of 70° and 80°F develop normal skin colour, odour is prominent, fruits are marketable but the

TABLE 1. *Effect of temperature on development of chilling tnjury in mangoes*

Storage temperature (°F)	Chilling injury (%)	Green firm (%)	Symptoms	Remarks after 4 days at 80°F
			After 22 days storage	
35	80	98	Lesion coalesced to form light brown patches.	Diseased, unmarketable
40	80	95	Lesions coalesced to form a few patches	— do —
45	80	95	Lesions scattered to form patches	— do —
50	50	90	A few lesions scattered	— do —
55	Nil	65	Looks normal, colour break	Unmarketable soft, lacks odour
			After 12 days storage	
70	Nil	60	Colour break from green to yellow	Marketable, soft, odourless prominent
80	Nil	30	Yellow to orange	Marketable, soft, edible ripe, good odour

TABLE 2. *Effect of temperature on the chemical composition of mangoes*

Storage temperature (°F)	After 22 days of storage			
	Total sugars %	Acidity %	Sugar/acid	Total carotenoids (μg % 100 g.)
35	12.0	2.68	4.4	424
40	13.0	2.64	4.9	435
45	13.1	2.56	5.1	491
50	15 3	1.57	9.7	1643
55	16.0	0.85	18.8	2432
70*	16.0	0 52	30.7	4662
80**	17.3	0.16	108 0	8937

*After 15 days and ** after 12 days

storage life is reduced (Table 1). Chemical constituents of the fruits stored for 22 days or less at different temperatures indicate an increasing trend of sugars, decline in acidity, increase in sugar : acid ratio and carotene content with raise in storage temperature from 50°F and these changes are minimum below 50°F (Table 2). Based on these studies, a temperature of 55°F was provisionally selected for intensive investigations on mango, since there was no obvious physical damage to the fruit although, the eating quality and consumer preference were impaired. Studies on respiratory drifts of fruits stored for 22 days or more at 55°F indicate slow and steady rate of respiration. A sudden spurt in respiratory rate is noticed when removed to 80°F which gradually declines, as the fruit tends to become soft (Fig. 1). Development of carotenoids in the mesocarp is markedly reduced in fruits stored at 55°F for 22 days and ripened at 80°F (Fig. 2) compared to the fruits stored and ripened at 80°F (Fig. 3).

Pre- and post-storage treatments[8] were tried in order to improve ripening quality and also to reduce spoilage. Fruits were removed from storage at intervals

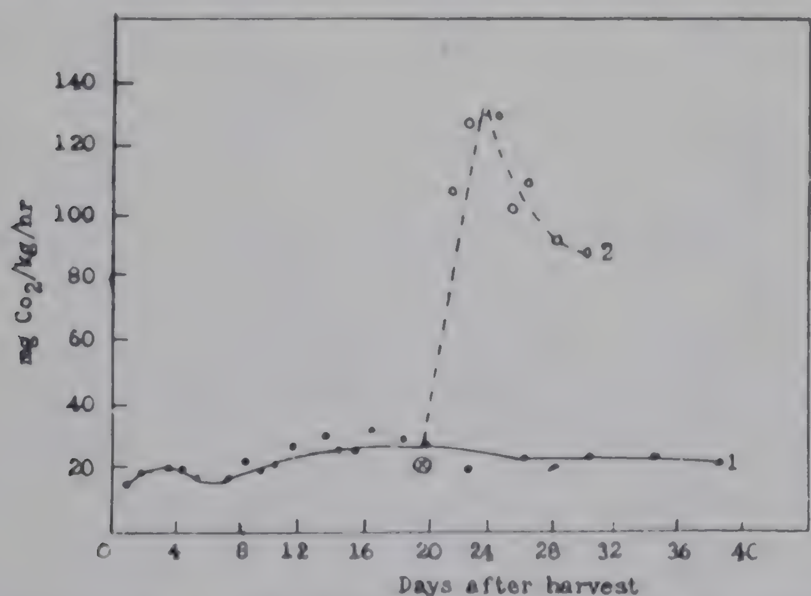

FIG. 1. *Respiratory pattern of mango during storage.* 1. *at 55°F.* 2. *On the 20th day frnits were removed and stored at 80°F for ripening*

of 6 days and warmed by dip treatment in hot water at 126°F for 5 min. This physical treatment reduced the spoilage due to fungi but did not improve the ripening qualities in fruits (Table 3). Pre-cooling and treatment of fruits with chemicals prior to storage have been found to be useful in improving the market quality of temperate fruits[1,9]. Therefore, chemicals such as diphenylamine, dimethylsulphoxide,

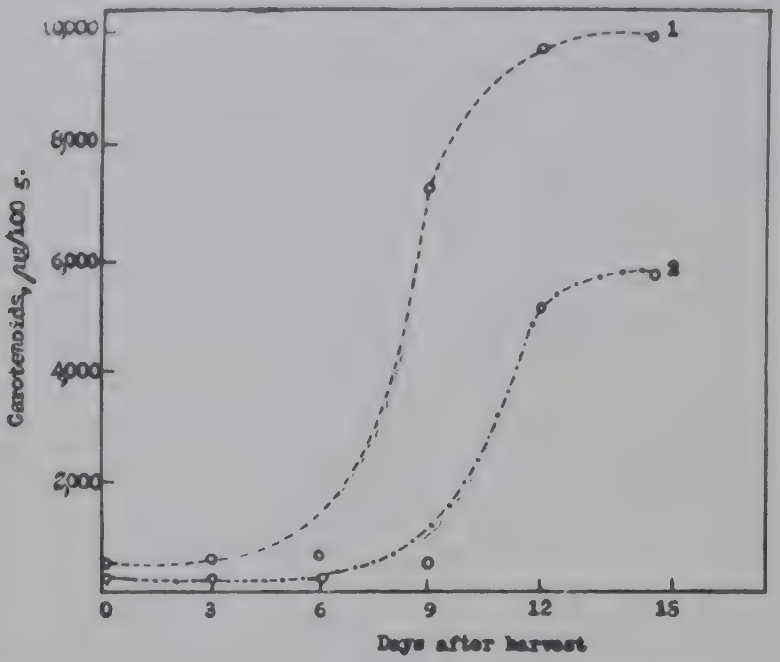

FIG. 2. *Development of carotenoids in Alphonso mangoes during storage at* 55°F. 1. *Total caroteuoids* 2. *β - - Carotene*

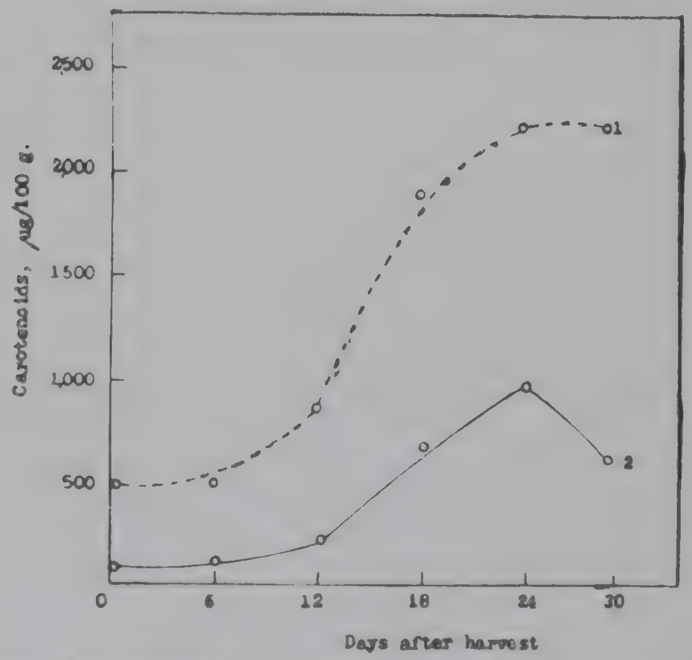

FIG 3. *Development of carotenoids in Alphonso mangoes during rtpening at* 80°F. 1. *Total Carotenoids* 2. *β — Carotene*

ethylenediamine tetra acetic acid and fungicides were included in chilled water at 50°F and fruits were hydro-cooled for 15 min. prior to storage at 55°F. Fruits were removed after 22 days of storage and 50% of the lot under each treatment was

TABLE 3. *Effect of post storage treatment on the incidence of low temperature breakdown in mangoes*

Storage period days	Ripening period days	Untreated		Hot water dip for 5 min at 126°F	
		Soft %	Spoilage %	Soft %	Spoilage %
6	6	32	9	33	5
12	6	74	39	78	11
18	6	74	43	80	17
24	6	87	81	88	29
30	6	93	94	91	76
—	16	78	39	87	15

Storage and ripening temperatures are 55 and 80°F respectively.

TABLE 4. *Effect of pre- and post-storage treatments on the incidence of low temperature breakdown in mangoes*

Pre-storage treatment at 50°F	After 22 days storage at 55°F			
	4 days at 68°F Treated prior to storage		4 days at 80°F Treated after storage in hot water (126°F, 5 min)	
	Soft (%)	Spoilage (%)	Soft (%)	Spoilage (%)
Hydrocooling	33	41	100	50
Diphenylamine 2500 ppm	33	36	89	41
Dimethyl sulphoxide, 2500 ppm	89	56	100	64
EDTA, 2500 ppm	35	60	100	74
Thiram, 2500 ppm	67	39	100	49
Captan, 5000 ppm	57	34	99	45
Zineb, 2500 ppm	77	42	100	64
Ziram, 2500 ppm	63	42	94	60

TABLE 5. *Effect of pre- and post-storage treatments on the incidence of low temperature breakdown in mangoes*

Pre-storage aqueous dip treatment at 77°F	After 24 days storage at 55°F and 6 days at 80 F for ripening			
	Treated prior to storage		Treated after storage in hot water (126°F, 5 min.) containing Ethephon (500 ppm)	
	Soft %	Spoilage %	Soft %	Spoilage %
Aqueous dip	85	59	88	29
Captan 5000 ppm	80	33	81	15
Aureofungin 500 ppm	90	59	91	23

warmed by dip treatment in hot water at 126°F for 5 min. These pre- and post-storage treatments did not improve the ripening characters and reduce spoilage in fruits (Table 4), although post-storage treatment stimulated softening process.

In a separate experiment, fruits were treated with fungicides like Captan and Aureofungin prior to storage at 55°F. Fruits were removed from storage after 24 days and half the lot under each treatment was held at 80°F for ripening and the other half was warmed by dip treatment in hot water containing Ethephon to stimulate ripening and subsequently held at 80°F for ripening (Table 5). Post storage treatment reduced the spoilage considerably but did not improve the quality of fruits with regard to odour and colour development.

Discussion

It is seen from the above studies that storage of mangoes at temperatures below 55°F results in visible physical damage on the surface of the fruit, although such injuries are not noticed at 55°F and above. Such fruits, when ripened lack characteristic odour, colour and taste, as a result of low temperature breakdown.

Several authors have indicated possible mechanisms of low temperature breakdown in fruits and vegetables[2,3-6,8]. One of the hypothesis is the accumulation of the toxin with temperature as advanced by Plank in 1941 (quoted by Fidler). Accumulation of aldehydes and alcohols as end products of disrupted metabolism was observed in mangoes stored at 55°F and in controlled atmospheric conditions[5].

Forced air circulation in storage rooms to flush out volatile emanations have also not been successful in improving the ripening qualities in mangoes[10]. An irrepairable change in protoplasmic vtscosity, or in the permeability of the cellular or mitochondrial membrane has also been demonstrated in several plant tissues subjected to chilling temperatures. Perhaps the earlier workers have failed to recognise this irrepairable damage caused to fruits as a consequence of low temperature storage but reconciled with an optimum temperature at which no observable physical tissue damage was seen. Detailed investigations on biophysical and biochemical aspects are essential to resolve this problem of LTB in plant tissues.

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Refrigeration Requirements for Radurized Sea Foods

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Sea foods are highly perishable products among the foods, temperature being the most important factor influencing quality and shelf-life. An increase of only a few degrees in the vicinity of 0°C considerably accelerates the spoilage. Thus, a reduction of temperature from 3° to 0°C reduces the the spoilage rate by 50 per cent and a reduction from 10° to 0°C lowers it by a factor[1] of 5 to 6. Hence, it has been a common practice to transport the sea foods packed in ice both in the fishing vessels and on land.

Nearly 70 per cent of the total catch in India is disposed unprocessed. However, limitation in the conventional preservation methods and non-existence of adequate net work of cold storage and refrigerated transportation facilities, have restricted the distribution of fresh fishery products in the country to a narrow coastal belt resulting in under-utilization of the resources and low *per capita* consumption. In the absence of alternate technology for processing mixed catches of fresh fishes and with the existing prohibitive costs of conventional methods for the preservation of commercially important fishes, radiation process as an effective alternate may offer a greater scope in achieving wider distribution and efficient utilization of fresh fishery products, including trash fish.

Radurization in the dose range of 100 to 200 Krad reduces total bacterial count by 1 to 3 log cycles[2,3]. However, since microorganisms differ greatly in their radiation sensitivity, a qualitative shift occurs in the micro-flora of radurized sea foods. The most active fish spoilage organisms, *Pseudomonas*, *Achromobacter* and *Proteus*—the gram negative psychrophylic bacteria—are relatively more sensitive, and hence, are selectively inactivated leaving behind biochemically less active mesophilic bacteria[3,4]. The combination of radurization, which reduces the number of psychrophilic bacteria and refrigeration temperatures, which limit the growth of mesophilic and thermophilic microorganisms while allowing the proliferation of psychrophilic species, therefore, has been found to be effective in extending the shelf-life of sea foods by 2 to 3 fold[4]. This would facilitate the distribution of fresh sea foods to the hinterland. This would also help a great deal in diversification in utilization of trash fish by providing almost 3 weeks[5] for converting them into secondary products such as fish cake, fish *kheema*, fish steak and fish finger.

Temperature of fish stored and transported in ice is seldom as low as 0°C after unloading from the vessel[1]. Investigations at Torry Research Centre[6,7] have revealed

that at the retail end of the distribution chain, over 20 per cent of samples were above 15.6°C, some showing temperature as high as 25°C. Initial bacterial counts being quite important in the subsequent rate of spoilage, radurization at early stages would bring down the spoilage rate considerably by reducing the initial bacterial load. This, however, does not minimise the importance of refrigeration temperature for subsequent storage and transportation since storage temperature directly affects shelf-life of irradiated sea foods generally to the same extent as that of unirradiated ones[1].

Some problems with respect to proliferation of pathogens, particularly *Clostridium botulinum* type E, may arise due to shift in flora with the use of sub-sterilizing doses of radiation.

Possibly, the simple and safe method to eliminate botulinum hazard appears to be temperature control. If the temperature of the sea food product is maintained below 3.3°C throughout the storage, transportation and distribution, botulism can be avoided. However, as earlier mentioned, the commercial samples seldom reach a temperature near about 3°C even if they are iced on the board itself[6,7]. Temperatures as high as 25°C observed in iced fish by Torry Research Centre point out to the difficult problem in maintaining low temperatures using ice, and hence, to the possible botulinum hazard in radurized fish under such conditions. These considerations show that refrigeration requirements for radurized sea foods are more exacting and stress the need for better refrigeration methods capable of continuous monitoring of temperature throughout the storage, transportation and distribution period. This problem has a greater significance in India due to prevailing tropical climate, temperature variations from place to place and improper icing followed by the usually untrained fisheries personnel.

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Physical and Rheological Properties of Perishable Products

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The processing interests of the agricultural engineer focus upon operations and physical properties related to handling, drying, separating, sorting, conditioning, treating, storing and marketing of agricultural products. The materials he handles, however, are generally not homogeneous and may be subject to rapid biological change prior to and immediately after harvest.

Physical Properties

The physical characteristics of perishable products are those which describe quantitatively or qualitatively the physical condition of the product. The shape, size, volume, specific gravity, bulk density, appearance, colour, surface area are some of these characteristics. These may be considered as engineering parameters of the product.

In order to describe the object, shape and size-are generally necessary though both are inseparable properties. In order to describe completely these properties with reference to a perishable product, infinite number of measurements are required.

Density and specific gravity of perishable products have many applications such as : separation and grading[8], maturity evaluation [9] texture and softness of fruits[9] estimation of air space in plant tissues[5] and quality evaluation of products such as peas, sweet corn, lima beans and potatoes which increase their density as they mature[7,1]. Because of the irregular shape and porous nature of biomaterial, difficulty is encountered in the determination of these properties. However, some techniques have been established such as platform scale, specific gravity, gradient tube, air comparision pycnometer, radiation method and porosity measurement apparatus.

In the handling and processing of fruits and vegetables, knowledge of surface area is important. Surface areas of fruits are important to know the spray coverage, respiration rate, light reflectance, colour evaluation and heat transfer studies.

Rheological Properties

The physical characteristic such as texture quality does play an important role in processing. The descriptive terms such as hardness, softness, firmness, stickiness, gumminess, fibrousness, chewiness and juiciness are concerned with texture. However, no exact definition of the word 'texture' as related to food has yet been offered. The biological tissues in solid food materials are composite in nature and are made up of

liquid and solid substances that are combined in a random fashion.

"Rheology is a science devoted to the study of deformation and flow in the material. By definition, time dependent mechanical behaviour is termed as rheological behaviour. Rheology, therefore, considers force, deformation and time such as time dependent stress and strain behaviour, creep, stress relaxation and viscosity[12]."

The perishable products frequently undergo physical changes. During development and storage the cells are influenced by external factors such as temperature, .humidity, oxygen, food supply and energy consumption. Because of this complex system in the study of rheology of biomaterials, emperical approach is preferred.

The tissues and fluids in food materials are capable of sustaining and recovering from a large deformation. For solids, stress-stain relationship is non-linear and liquids deviate from Newtonian flow behaviour. In general, biomaterial of agricultural origin are viscoelastic and this behaviour is non-linear. The initial part of the force-deformation curves of soft biomaterial are usually concaved towards the force axis and that of dry polymeric material convexed towards force axis.

To evaluate an elastic modulus for the flesh of fruits and vegetables, the conventional test of cylindrical specimens of the material, subjected to uniaxial compression between two parallel rigid plates have been rcported for apple[13] potato[15] and cheese and butter[4].

The shearing strength which is another important characteristic measured, decreases as the fruit ripens and since the pectic substances decompose the connection between cells weakens. In an unripe fruit, the cells along the shearing surface are tightly held together and tear off in response to shear stress.

It has been indicated that none of the biomaterial tested so far show perfect elasticity. Loading and unloading of biomaterial for several cycles show reduction in plastic deformation. If loading and unloading cycle results in a closed loop, the behaviour is called elastic hysterisis.

Stress relaxation is a rheological behaviour in which a material is subjected to a constant strain and the decay of stress as a function of time is recorded. Where as in the creep behaviour, material is suddenly subjected to a dead load and deformation with time is recorded. Several methods are available to determine these behaviours.

So far only little work has been done on determination of the physical and rheological properties of biomaterial. Due weightage is essential for further study.

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Some Reliable Parameters for Assessing the Quality of Frozen Fish

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The rate of freezing has long been considered of vital importance for obtaining high quality frozen fish products. Frozen fish deteriorate in quality after storage for several months due to changes in texture. Taste panel gradings have been followed to determine changes in texture of fish during frozen storage.[1] Relationship between texture of the flesh and the state of muscle proteins is evidenced by the fact that they become less easily extractable by dilute salt solutions on storage in the frozen state for some time.[2-8] Fish muscle seems to differ most characteristically from mammalian muscle only in its very low content of connective tissue and also in its albumin content.[9-11] During frozen storage adverse texture changes—an indication of denaturation. is confined to the structural muscle fibre actomyosin—the protein mainly responsible for texture of fish muscle.

Actomyosin from fresh fish muscle could be extracted by blending it with 5% NaCl in 0.02 M $NaHCO_3$ at pH 7-7.5 (μ =0.875). Actomyosin is also determined by precipitating this total salt soluble protein extract by diluting it with nine volumes of chilled distilled water (μ =0.087).

Although actomyosin denaturation has been measured by various methods, the criterion for denaturation resulting in texture is the loss of extractability.[12] The mechanism by which actomyosin becomes insoluble, is yet not well understood.[13] Toughness may also result from changes in cell membranes. During protein denaturation either intercellular (inter fibrillar) linking or cementing together of larger structural components or intercellular cross linking of the myofibrillar proteins at near the molecular level is implicated.

Denaturation of fish muscle protein at low temperature is very closely linked to the release of free fatty acid (FFA) from phospholipids. There may be a relation-

ship between lipid hydrolysis and actomyosin denaturation in frozen muscle. Many workers[14-17] have reported a correlation between increase in the fatty acid content and decrease in actomyosin extractibility in frozen cod muscle. These reactions can result in protein binding and aggregation under conditions encountered in the frozen tissue and contribute to the deterioration. Protein solubility falls to a different level for different species of fish.[18] Neutral lipids present in sufficient quantities seem to protect the proteins from denaturation by FFA. It is possible that the concentrated tissue salts foster denaturation by providing a favourable environment for the reaction and that it is actually the FFA which directly cause denaturation.

The cell fragility measurement (E $\frac{1}{2}$ cm) always falls during frozen storage of fish.[19] It is based on the observation that after being thawed, the individual cells become progressively more difficult to rupture with homogenizers as storage time increases; optical density of the homogenate in dil. formaldehyde solution is measured in a colorimeter. Although the rates of denaturation of frozen fish as measured by change in soluble protein and cell fragility are not the same,[20] cell fragility measurement changes are more rapid than the taste panel assessment. A high cell fragility reading (E $\frac{1}{2}$ cm) may represent a tough fish of low pH or a fresh cold stored fish of high pH which is highly acceptable to the palate and not denatured by cold storage.

Miscellaneous Studies

The classical molecular model[21] of muscle has been expanded by Ebashi's description of α-actinin and β-actinin. The length of an F-actin filament *in vivo* is considered to be the result of a balance between the polymerizing effect of α-actinin and the depolymerizing effect of β-actinin. It is conceivable that the Z-band material which is altered during freezing and thawing is manifestation of this balance *in vivo*.

There is a need to determine whether or not the fish has been frozen and thawed. The earlier techniques emphasised on the dehydrogenase activity of the muscle tissue and the microscopic examination of the red blood corpuscles and more recently the malic enzyme activity.[22] A method of colorimetric[23] measuring of fading of astaxanthonin pigment of the shrimp during frozen storage has also been suggested.

Enzyme activation occurs at temperature just below the freezing point. Release of hydrolytic and other enzymes from the various cells causes damage to lipid protein membranes during thawing or during storage at relatively high frozen temperature where even a small temperature fluctuation may have a large effect. Lipid, nucleotide and protein component, may be attacked and thus affect denaturation of protein.

Microbiological Aspects

The bacterial load in the frozen product is partly dependent on the initial load of the raw product prior to freezing and is primarily dependent on its treatment and sanitary conditions. Microbiological requirements per g of frozen fish products are: Maximum total plate count at 37°C shall be 100,000/g. *Escherichia coli* faecal type, *Salmonella*, coagulase +ve *Staphylococcus* and other pathogenic bacteria shall be

absent.[24] To comply with the above, more precautionary measures have to be taken to ensure hygienic conditions. New sensitive methods for detecting and enumerating *Salmonella*, coagulase +ve *Staphylococcus* have been developed.[25,26] Also considerable importance is now being given to pathogenic organism *Vibrio parahaemolyticus*.[27]

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Some Aspects of Chilling Fresh Water Fish for Marketing in Fresh State

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Reactions leading to spoilage commence almost immediately after harvest in the case of fish and hence, unlike in the case of other perishables, measures necessary for preservation can not be delayed. Chilling is probably the only practical way to retard these reactions and also to hold back bacterial attack which accompanies the biochemical changes and to retain freshness in fish. Chilling, though a short term preservation, is of enormous commercial importance in view of the fact that a sizeable proportion of people throughout the world still buy fresh fish in preference to frozen or processed fish. This is evident from the fact that about 20 million tons out of a total world catch of 70 million tons of fish are marketed fresh annually[1]. In India, the proportion of fish utilized in the fresh state is still higher (about 69% of the total). There has been in recent years a corresponding decline in production of cured and dried products, which was obviously the only way to meet the seasonal and local glut. With improvement in availability of ice at producing centres, more and more of fresh fish are being chennellized into interior markets every year.

Place of Fresh Water Species in Indian Fish Trade

A second and more probable reason for the larger preference for fresh fish in India is the dominant share held by fresh water species in our fisheries. Nearly a third of fish produced here belong to inland species which are exclusively utilized in the fresh state. There has been a steep rise in fresh water fish production during the past few years (from 477.5 thousand tons in 1966 to 690.2 thousand tons in 1971)[1]. Many of them,especially the major carps, are varieties having preferential markets, justifying long distance transportation. A survey conducted reveals that about 80% of fish carried by Indian Railways belong to fresh water species[2]. This fact deserves special attention while considering improvements in the existing system of marketing, because a thorough knowledge of the physico-chemical and bacteriological aspects of fresh water fish spoilage is essential for this purpose. Chilling of fish near freezing point can bring about significant changes in the storage life of sea fishes[3]. Temperature influences the spoilage rate in two ways (*a*) lowering of temperature prolongs the duration of rigor mortis. It also retards all those other reactions including proteolysis brought about by muscle enzymes, (*b*) more important is the role of temperature on bacterial growth. Apart from increasing the lag phase, quick reduction in body temperature drastically brings down the growth rate of bacteria.

Thus from the refrigeration angle the two important requirements for efficient

chilling are (*i*) removal of heat as rapidly as possible from the body of the fish immediately after death, to produce a longer rigor period, to retard autolysis and to increase the lag phase of bacteria, (*ii*) bringing the temperature as close to freezing point as possible to reduce the growth rate of spoilage bacteria. Care should be taken that fish does not get frozen because it is well known that freezing just below 0°C produces undesirable textural changes. Other important factor associated with the process of chilling is ambient humidity in relation to the medium of chilling. Modifications of ambient conditions like use of CO_2 atmosphere, use of preservatives like antibiotics, irradiation, etc. have peen used along with refrigeration with varying degree of success.

Advantages like having high specific heat and heat of fusion have helped ice to retain its position as the best chilling medium for fish, especially during transportation. Moreover, melting ice forms a heterogeneous medium, effecting better heat transfer. Salt water ice, by virtue of lower melting point, can be used with definite advantage.[4] In large scale handling, the usefulness of chilled water or brine cannot be overlooked. Possibility of circulation or spraying, and ability for faster cooling by completely surrounding the object are the merits of a liquid medium[5]. Though air chilling is comparatively slower and requires controlled humidity conditions to prevent dehydration, the economics of using mechanically refrigerated transport systems especially in long distance movement of fish have to be considered.

Present Status

At present, the railways account for about 50,000 tons of fish transported annually from major producing centres to important urban markets[6]. Except a few refrigerated wagons running between few stations, all fish transported by rail travel in uninsulated ordinary parcel wagons. The period of transit between consigning and terminal stations vary between 24 and 60 hours[7] The fish is packed in wooden boxes or bamboo or reed baskets. Insulation is effected by a lining of dry leaves inside or a cover of gunny cloth outside. The fish is mixed with ice varying in proportion between 40 and 100% of the weight of fish depending on the season, length of journey etc. Re-icing is done in long distance transport at midway stations.

The refrigerated wagon used by the railways is made up of two chambers separated by a smaller compartment housing compressors and control panels.[7] Each chamber has a capacity of 7½ tons, and are cooled by blowers mounted on expansion coils. The compressor is powered by generators driven during the movement of the train. The wagon is usually hauled by fast trains while carrying pay load. Pre-cooling is done before loading to —2°C. The temperature of the chambers during journey varies between —1°C and 4°C. Data are not available on the humidity of the chambers. The consigners usually mix some ice with the fish packed in ordinary containers detailed above. These wagons carry nearly 5000 tons of fish per year.

Bulk of the fish moved by road transport is done in open uninsulated trucks. Refrigerated and insulated road vans are few and quantity moved by them is insignificant.[8]

The survey showed that only about 20% of the fish arriving at destinations

remained below 5°C. About 70% of recorded temperatures varied between 5°C and 20°C. About 10% of them rose above 20°C. In most cases complete melting of ice accounted for the rise in temperature. In a few cases, block formation of ice at the top and consequent lack of contact with fish prevented efficient heat transfer. With hot air from outside entering the inner cavity between ice and fish, the temperature of fish rose in spite of presence of ice.

The quality of the fish truly reflected the variations in temperature. Data collected at Howrah by the Central Fisheries Corporation for a period of three months preceding May 1969 show that 53.1% of fish marketed was substandard when the period of journey involved was one day, and 81.3% was substandard when the period was 5 days. Observations of consignments arriving at Howrah during the same period showed more or less similar pattern of spoilage. It may be added that substandard fish sells at 50% of the prevailing market rate for standard fish. This is a substantial loss in view of the fact that 30,000 tons of fish carried by rail is sent to Calcutta alone.

Some Laboratory Observations

With the above background, some basic studies were carried out in this laboratory regarding the spoilage characteristics of a few commercially important fresh water species i.e. major carps (Catla, Rohu, Kalbasu, Mrigal) during chilled storage.[9]

These were followed by studies on the influence of some factors involved in handling and transport on the quality of the fish.[10] An important observation was that in ideally iced condition, these varieties possessed remarkably longer shelf life as compared to most marine species. Judged by organoleptic, chemical and bacteriological standards, they remained acceptable for nearly 5 weeks, whereas the reported storage life of comparable marine species under similar conditions ranged between 9 and 21 days. The growth rate of bacteria suggested the presence of a flora different from those of marine environments. The influence of variations in temperature on spoilage ratio also was unique. Cod, a typical marine white fish, is reported to have become unacceptable at the end of 3, 5, 6, 11 and 16 days when stored at 4, 2, 1, —1.25 and —2°C respectively.[11] But in the present case, Mrigal, a member of the major carps, remained acceptable for 12, 18 and 35 days at 10, 5 and 0°C respectively. The former typifies the large influence of temperature on marine fish spoilage illustrating growth pattern of marine *Psychrophiles* at lower temperature. The latter presents a different picture showing only sluggish variations in spoilage rate with comparatively wider temperature fluctuations. It was also observed that species and size differences did not exert much influence on quality changes in storage. However, this situation will be different if the fish contain[12] large amount of feed in the intestines. When stored in a chill room un-iced, low humidity (below 50% R.H.) caused dehydration and shrinkage resulting in loss of weight and bad appearance. On the other hand, highly humid atmosphere (above 90% R.H.) was found to create a favourable condition for fast growth of surface bacteria. Some loss of flavour is observed when fish is stored in melting ice or chilled water, presumably due to leaching.

The practice of marketing fish as frozen fillets is gaining popularity particularly in urban areas. A system of transporting and distributing chilled fillets will have additional advantages. But there are many problems associated with packaging and storage. For instance, leaching and consequent loss of flavour and other solubles preclude the possibility of using ice directly for chilling fillets, necessitating a water proof system of packaging. This leads to the need for standardizing conditions in relation to packaging materials, mode of chilling and associated bacteriological studies.

The domestic fish trade is not very much organized at present and the practices followed are traditional and quite inadequate for a commodity which needs specialized attention. Well defined general codes of handling practice for fish are available to the industry, but any improvement to be effected in a given set of conditions necessarily calls for an understanding of the problem in relation to the nature of the raw material, environmental factors and resources available. The account presented above is part of the efforts made in this Institute in this direction.

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A Study of the Changes in the Flavour Components of Fish and Fish Fillets Related to the Nucleotides during Ice and Frozen Storage

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Fresh water fish is of great economic importance in countries like India and increased production on a substantial scale is envisaged in the coming years. Flavour is the dominant quality factor deciding consumer appeal and is closely related to

handling and storage subsequent to catch. A recent survey by the C.F.T.R.I. has shown that 25 to 50% of the fish reach the main markets of West Bengal in sub-standard condition. Refrigeration is the method of choice for the short-term storage and transport from catch to the fresh fish markets. Nair *et al*[1]., have reported loss of flavour even under controlled laboratory conditions of refrigerated storage. Flavour of fresh fish is of complex nature but has been shown to be dependent to a great extent on degradative changes in the nucleotide fraction brought about by enzymatic and microbial reactions. A systematic study of the distribution of the nucleotide compounds in fresh, ice-stored and deep frozen fresh water fish is reported in this paper.

Experimental

Barbus carnaticus, one of the important commercial varieties of locally available carps has been employed in the present investigation. For the study of ice storage in case of whole fish (without evisceration or dressing) fish of the size 22″ to 25″ in length and weighing 4 kg. each were caught in Cauvery river and transported in ice to the laboratory. Initial sampling was performed on the spot from two fish and the others (6 nos.) which were transported quickly to the laboratory were iced in 1 : 4 ratio in an insulated box and stored at 5 to 6°C. For the second experiment dealing with ice stored fillets fish of a smaller size 10″ to 12″ and weighing 2 kg. each were picked up from fish ponds maintained by the State fisheries' authorities. Fillets of size 3″ × 3″ × 0.7″ were separated from the muscle after cleaning and evisceration and packed in polythene bags which were buried in ice and stored at 5 to 6°C. Fish were sampled on 5th, 10th and 15th day for determination of nucleotides and related compounds. For the study of frozen fish, four of the larger fish (22″–24″) used for ice storage were frozen in a laboratory deep freezer at −20°C after evisceration and stored for 3 months at the same temperature before analysis.

Estimation of nucleotide compounds and related substances was performed after perchloric acid extraction, according to Jones and Murray[2]. Paper chromatography was employed for separation and identification and quantitative estimation carried out by spectrophotometry with Beckman Model DU, the spots being eluated with N/100 HCl to give the optical density in UV light. Results of the above study are presented in Figs 1 and 2 and Table 1.

Discussion

The object of the present investigation is to study the post mortem degradation of nucleotide compounds in fresh water fish since inosine monophosphate (IMP) has been identified in recent years as the major flavour enhancing component of animal foods and study of its degradation to hypoxanthine has heen advocated as the most reliable index of fish spoilage. Apart from the paucity of information regarding the IMP content of fresh water fish, it has been observed that post mortem degradation and deamination of ATP to IMP is much faster in the case of tropical fish.[3] In fact hypoxanthine itself is detected in freshly landed condition in tropical fish and shell fish much more earlier than in temperate waters.

IMP constitutes the main nucleotide component (2.0 μM/g) in a fresh water

TABLE 1. *Changes in nucleotides and their derivatives before and after frozen storage (—20°C) for 3 months in Barbus carnaticus (μM/g)*

Compound		Before	After
Adenosine-diphosphate (ADP)		0.89	0.05
Adenosine-monophosphate (AMP)		0.03	0.03
Inosine-monophosphate (IMP)		2.01	2.92
Inosine (INO)		0.42	0.21
Hypoxanthine		0.56	1.99
Adenosine (ADO)		1.75	0.52
Guanosine-monophosphate (GMP)		0.66	trace
Guanine		—	0.43
	Total	6.32	6.16

fish like *Barbus carnaticus*, in freshly caught fish followed by ADP with AMP occurring in traces. Persistent presence of small amounts of ADP in fish during prolonged ice storage has been noted by Spinelli[4]. It is also partly bound to

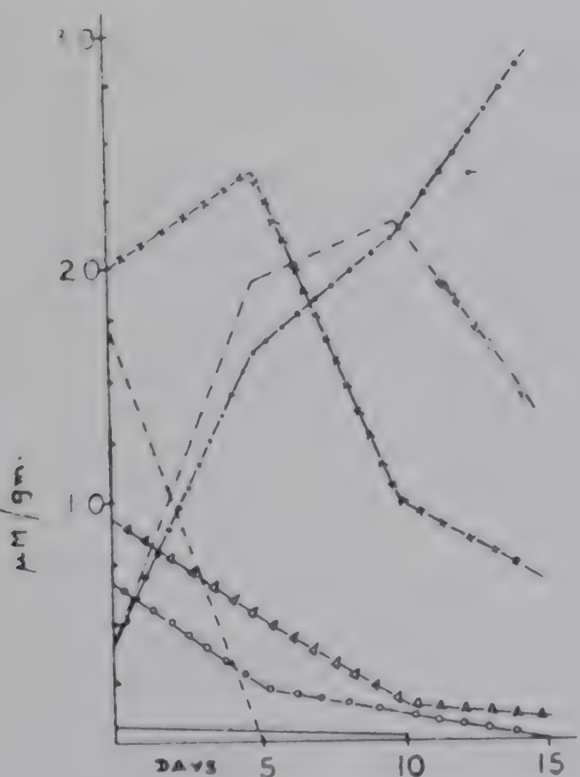

FIG. 1. Changes in nucleotides and their derivatives during the ice storage of whole fish.
·———· AMP; Δ———Δ ADP; ×———× IMP;
0——0 GMP. ——— ADO; —.—.—. INO; .———. HYPX

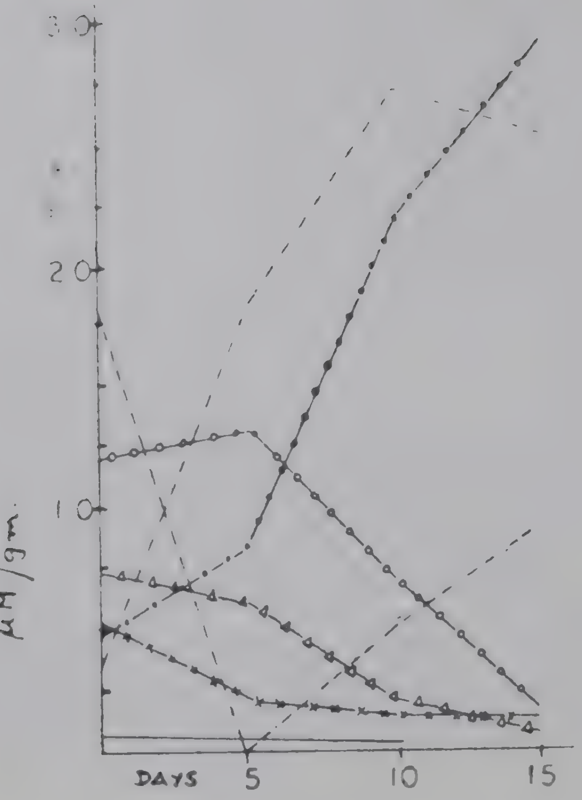

FIG. 2. Changes in nucleotides and their derivatives in the ice storage of fish fillets. .———. AMP; △————△ ADP; O————O IMP; ×————× GMP. ———— ADO; —.—.— INO; ·————· HYPX; ——.—— Uric acid.

myofibrils and rendered inaccessible to enzymatic action in initial stages[5]. To account for the absence of AMP in the present samples, one can postulate a separate ADP-deaminase system as pointed out by Webster[6]. IMP is quite stable and shows actually some increase in the case of whole fish upto 5 days after which rapid degradation occurs during the 5th to 10th day. Loss of flavour in the earlier period can however be accounted for by the rapid rate of hypoxanthine production during this period. Moreover GMP which is even superior to IMP in flavour enhancing property has also been observed in these local carps and its level has registered a rapid decrease as seen in Fig. 1. Rapid increase in INO level during the first 5 days finds a parallel in a similar decrease of ADO during the same period indicating that ATP degradation via AMP has an alternative route through ADO and its deamination as observed by Tarr and Comer[7] in the case of lingcod. After 5 days hypoxanthine alone shows a steady increase. The present ice storage study has revealed that breakdown or loss of IMP which may influence the flavour is slower than in the most marine fish[8,9] with the exception of sword fish[10].

Changes observed during the ice storage of fillets show a similar pattern in general, although they are not strictly comparable because of obvious reasons. Fish employed for the study of fillets were of a smaller size which might account for the lower level of initial IMP concentration. There is a popular belief dating back to Chancer that the flavour of fish is related to age and size, unlike the meat of land

animals and in view of the large variation in the size of any species of fish with age in case of fresh water fish, significant difference in the IMP content, if any, needs further study. Fall in IMP level is much more in evidence in the case of fillets. Although these have been packed in polythene, loss through drip formation is not unlikely but till the fifth day hypoxanthine is lower in fillets (0.84 μM/g) as compared to the whole fish (1.64 μM/g). Increase in hypoxanthine is faster in the later stages of ice storage as compared to whole fish. Since nucleotide degradation becomes partly microbial beyond the stage of INO, the faster rate of formation of hypoxanthine indicates that bacterial spoilage appears to set in the case of fillets. Oxidative changes in particular may exceed those observed in whole fish as it may be observed that uric acid the final product of ATP degradation by oxidation of hypoxanthine through bacteria is clearly revealed in the case of fish fillets and not in whole fish. For snort term storage however filleting may have no disadvantages.

From Table 1 it may be observed that frozen fish show even after 3 months storage retention of IMP and in fact an increase (from 2.01 to 2.92 μM/g). Both ADP and GMP have undergone complete degradation while ADO is reduced by deamination. Hypoxanthine has registered considerable increase from 0.56 to 1.99 μM/g. An overall impression gained from the study based on the retention of IMP indicates that IMP breakdown proceeds at a slower rate compared to other enzymatic reactions which are evident even in frozen state.

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Importance of Refrigeration in the Modernisation Programme of Slaughter Houses

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The most important factor necessitating chilling of carcasses coming off the slaughter line in present day abattoirs is the inevitable presence of bacteriological

contamination on the surface[1]. The lymph glands are also an important foci of bacteria. In modern practice, the regulation of ante—and post-mortem veterinary examination is essential[2,3]. The various routes of bacterial contamination and hazards during the slaughter of animal, evisceration and dressing of carcass have been thoroughly reviewed by Ayers[4]. Washing of the surface of carcasses ready for chilling using hot water at 80-85°C has advantageous bacteriological implications[5].

The unavoidable presence of a small number of bacteria on the meat carcass is not of grave consequence under condition when the meat goes to the retail channel and cooking in the course of a few hours. The drop in pH of the meat during the development of rigor mortis[6] provides substantial resistance to bacterial proliferation. Also the culinary practices in our country involve thorough cooking in preference to methods like cooking rare and medium done. The mild cooking preferences and the consequent need of ageing the meat for tenderisation[7] in western countries dictate the necessity of dissipating the body heat, and chilling and storage of the carcasses in refrigerated rooms.

The surface temperature of the ready carcass at the end of the slaughter line would be same as the ambient temperature but deep inside the tissue the temperature would be somewhat higher than the body temperature of the live animal. So long as the animal is living, the bacteria in the lymph glands are under the restraint of the physiological defences of the animal. But once the animal has been slaughtered and dressed, these bacteria tend to migrate to the surrounding tissues and cause bone taint/souring. To prevent this type of internal spoilage, it is important that the carcass be cooled promptly. In the early days, this was achieved by hanging the carcasses in a well ventilated and screened room. The ventilation helps in reducing the carcass temperature and prevents build-up of humidity. But in this method of dissipating body heat of animal, there is a build-up of bacterial population from 10^3–10^4 to 10^6–10^7/sq.cm. and considerable loss of weight by dehydration in the range of 2.5-3.5%. The present practice is to load the carcasses from the slaughter floor into precooled chilled rooms to reduce the temperature of the carcass as quickly as possible without any surface freezing.

Air in the chill room is circulated at 2m/sec. and regular spacing of carcasses is maintained to ensure uniform chilling. The humidity of the chill room is maintained at 88-92%. The refrigeration capacity and the number of carcasses loaded into one chilling room are matched so that temperature of 4°C is attained in the deepest part of the carcass in 16 hours after loading and the temperature of the room is not allowed to increase beyond 4°C[8]. In very large beef carcass a longer duration is needed. The present practice is to have two refrigerated rooms, one to reduce the temperature of the carcass to 4°C and the other to hold the chilled carcasses for further ageing. The chilling room has the larger refrigerating capacity and arrangements to keep the air in circulation at the required speed. The chill rooms in some of the modern abattoirs also have elaborate arrangements to maintain the humidity of the circulating air at saturation to keep down the evaporative loss to 1-1.2% only[9]. Once the chilling phase has been achieved, the carcasses are transferred to the holding room.

The usual temperature on the evaporator coils of chill rooms are as low as

$-18°C$ so that the room temperature at the start of loading of warm carcasses is $-4°C$ and does not rise more than 3°C at any time. Attempts to bring down the temperature of the carcass at a faster rate led to undesirable effects. When a warm muscle is suddenly exposed to a very cold temperature, a phenomena known as cold shortening takes place[10]. Slow development of rigor mortis from a relaxed state is necessary for obtaining tender meat. In the case of beef carcasses shrouding in cloth wetted in warm saline water is advocated to smoothen and improve the appearance of the surface fat ; shrouding also reduces the shrinkage.

After chilling the carcasses are transferred to the holding room which is maintained at 3°C and 88-92 humidity. Beef carcasses are held for 4 to 7 days for ageing changes. Pork and lamb carcasses are held only for one day after chilling. The practice of holding the chilled carcasses is found to be necessary to give time for the rigor mortis changes in muscle contraction to get resolved and to initiate tenderisation. Since these tenderisation changes are temperature dependant, holding of the carcasses at a higher temperature of 42-60°F in the presence of UV irradiation (Tenderay process) or as cuts vacuum packed in heat-shrinkable Cryovac bags at 68°F (Cameroon process) have been commercialised in the U.S.A.[11] UV irradiation had vacuum packing in Cryovac preclude the development of organisms that form slime on the surface of meat. Cryovac packing also has the advantage of preventing shrinkage. Specialised installation of UV light and arrangements to keep the carcasses in movement are needed to ensure equal exposure of all surfaces in the case of Tenderay process. The fat on the carcasses may turn rancid if overexposure to UV takes place. After the desired tenderisation has taken place the meat has to be stored at usual refrigerated or frozen temperature.

Freezing and frozen storage facilities are usually not built along with the slaughter house unless the slaughter house is a component of the plant manufacturing meat products and/or caters to long distance transport. Freezing and transport of whole carcasses in the case of veal and lamb and quarters in the case of beef used to be popular. But, now freezing and storage in the form of prepacked cut is gaining in popularity. Quick freezing either by blast freezers in the case of irregular cuts or plate freezers in the case of pieces of uniform thickness is the favoured method rather than freezing in rooms with low frozen temperatures. This practice has become popular after it was recognised that large ice crystals are formed during slow freezing and these give rise to increased drip loss on thawing.[12] Rooms with low temperatures of the order of —18 to —25°C are used only for frozen storage of quick frozen meat.

Western practices like ageing of the carcasses have to be evaluated in relation to our cooking practices. In general, the cooking practices of meat in our country are always thorough cooking in contrast to western practices of 'well done', 'medium done' and 'done rare'. Our cooking practices more or less resemble stewing. The tenderising effect of cooking as practiced in our country is far greater than any tenderisation brought about by ageing due to holding the carcasses in cold rooms. Holding the chilled carcasses for ageing add to the overall cost to some extent. In cases, where the meat is sold and cooked in 6 to 8 hours, chilling of the carcasses meat may not be essential. Therefore, the cold storages to be built as an ancillary

to the slaughter houses need not have a capacity equal to the output of dressed carcasses from the slaughter floor. A smaller cold storage of a third or quarter the output may be sufficient. Hygiene and sanitation during the slaughter and handling and proper ante- and post-mortem examination should have priority over cold storage facilities during the modernization of slaughter houses. In order to utilise the glands for pharmaceutical preparation (particularly insulin) adequate frozen storage facility should be established at in all slaughter houses.

In all slaughter houses, existing as well as modernized thorough ante- and post-mortem examination of meat stock should become an essential feature. No doubt our thorough cooking practices afford marginal protection against some of the hygienic lapses during evisceration and dressing. However milder cooking methods will probably become more prevalent with the increase in tourism and simulation of western sophistication which will make scrupulous hygienic and sanitation procedures essential. These milder cooking methods require meat that has been tenderised by ageing. Proper facilities like chilling and holding rooms, have to be developed in the slaughter houses to cater to such specialised demands.

Summary

In a modern abattoir, facilities for chilling and holding of the carcass meat, edible and pharmaceutical by-products like liver and pancreas, cutting, deboning and packing of meat, preliminary salting of hides and skins, processing of guts into casings and rendering are built complementary to the slaughter hall.

In the context of the modernisation of abattoirs in our country, the primary objective should be proper ante- and post-mortem examination, hygienic and careful operation of the slaughter line and full utilisation of by-products. Cold storage facilities for prompt collection of pharmaceutical by-products should be available in all slaughter houses. Chilling and holding rooms for the carcass meat should be decided based on the pattern of distribution and retail sale and the need for ageing of the meat.

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Studies on Refrigerated Storage of Eggs

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Normally, it takes about a week for cold stored eggs to reach the consumer. It is reported that egg quality deteriorates quickly at room temperature after the eggs are taken out from cold storage. As such, this investigation was carried out to determine the effect of storage on egg quality in refrigerator at 40°F and R. H. 85% (average) as against room temperature of 70°F and R. H. 55% (average) and to find out the rate of quality deterioration of eggs after cold storage, when kept at room temperature. Egg quality evaluation for room storage was scheduled after 3 and 7 days following cold storage. Depending upon the season, the egg will lose its quality beyond 10 days storage at ambient temperature.

Materials and Methods

240 eggs from 8 months old commercial layers kept under identical conditions of feeding and management, were procured. 40 paper cartons containing 6 eggs in each were tied with rubber bands for storage studies. These cartons as well as the eggs were marked from 1 to 6 in each carton. Each egg was weighed. The cartons were arranged in the refrigerator and in room temperature for withdrawing and testing them periodically for quality as follows :

The criteria for the studies were (*a*) weight loss during storage, (*b*) height of yolk, (*c*) width of yolk, (*d*) yolk index, (*e*) height of albumen, (*f*) width of albumen, (*g*) albumen index, (*h*) Haugh unit and (*i*) pH of whole egg, and these were determined as per standard procedures.

TABLE 1 *Schedule of egg quality storage and evaluation*

Storage period (days)	No. of eggs evaluated at R.T.	No. of eggs evaluated at refri. temp.	Refri. & then No. of days at room temp. evaluated (days)	No. of eggs evaluated after refri. storage
0	24			
3	24			
7	24			
10	24	24		
13			3	24
17			7	24
21		24		
24			3	24
28			3	24

Results and Discussion

Weight loss : The percentage weight loss of eggs under refrigerated and room temperature and also refrigerated, followed by room temperature storage is presented in Fig. 1. Weight loss increased with the number of days of storage. Khusu and Haleem[1] observed a weight loss of 0.65, 0.76, 1.61, and 1.69 g in 3, 7, 10 and 21 days in refrigerated stored eggs. Eggs kept in air lock room for the above period lost 0.78, 0.76, 2.19, and 2.40 g. The eggs were collected from layers on the 9th and 12th months of laying period.

In this study, the weight loss in case of eggs stored under refrigerated condition at 40°F and 85% R. H. was less than that in other storage conditions. But the loss was found more when kept in room temperature after refrigerated storage. The percentage weight loss in case of refrigerated temperature was 1.25 and 2.77% at intervals of 10 and 21 days respectively while it was 2.37 and 4.22% respectively on 3rd and 7th day after 10 days of refrigerated storage. In the other batch of eggs after 21 days refrigerated storage egg quality on 3rd and 7th day at room temperature, the losses were found to be 3.75 and 6.12% respectively.

In case of room temperature, percentage weight losses on the 3rd, 7th and 10th

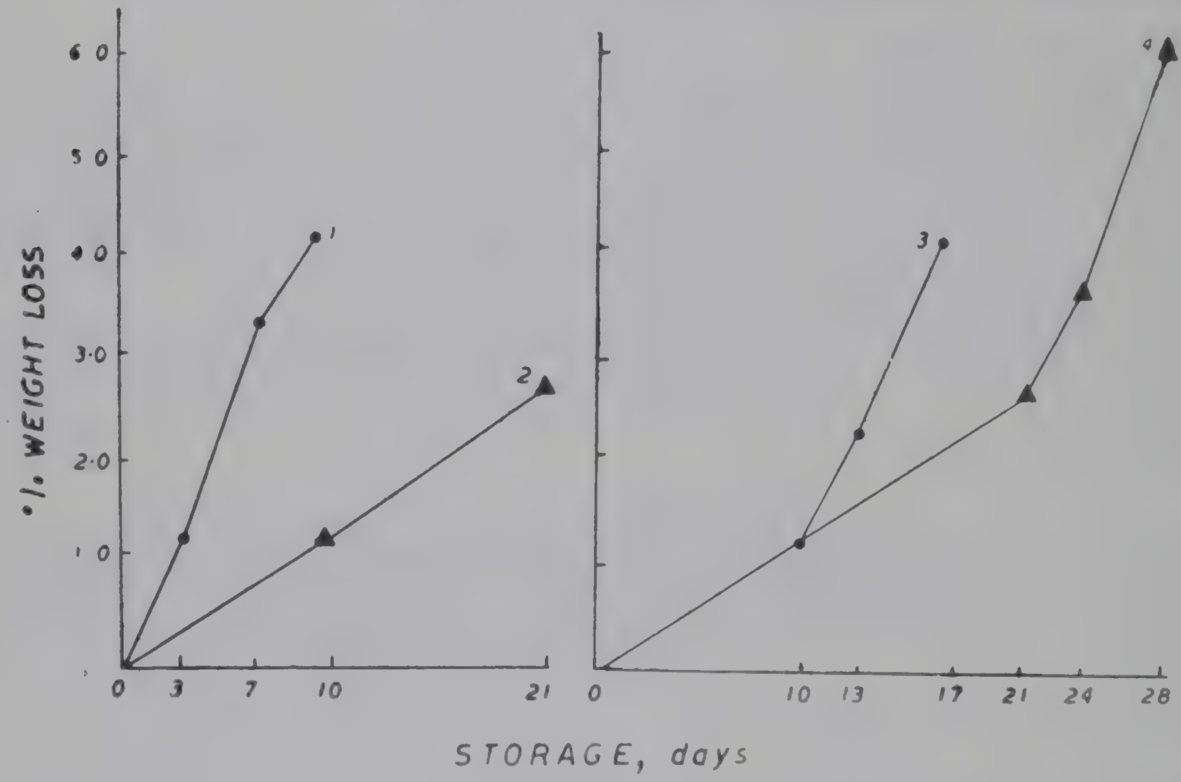

Fig. 1. Weight loss of eggs

1. Percent weight loss at room temperature
2. Percent weight loss at refrigerated temperature
3. Percent weight loss in 10 days at refrigerator and then keeping at room temperature for 3 and 7 days
4. Percent weight loss in 21 days at refrigerator and then keeping at room temperature for 3 and 7 days

day were found to be 1.27, 3.43 and 4.26% respectively. Therefore, the rate of loss in weight appears to be faster when eggs are kept in room temperature after refrigerated storage.

Vaurosok[2] reported the factors responsible for the deterioration of egg quality under storage. His results indicated that in high temperature and high humidity, the losses are more than at storage under normal conditions. Internal quality loss after storage at 82°F for 7 days or more have been reported by Mueller[3]. He concluded that relative humidity during storage which controls the rate of evaporation has little or no effect on albumen and yolk deterioration. Haleem[4] reported reduced egg weight, yolk weight and shell weight when Furazolidone was incorporated at increased level in diets of layers. In our studies, eggs stored at 70°F and 55% R. H. that is room temperature, the loss was more as compared to refrigerated temperature which is expected. Reddy and Panda[5] have reviewed the chemical changes occurring in shell eggs during storage. Fresh eggs during storage undergo many important changes such as, reduction in weight, development of stored flavour, reduction in nutritive value and finally thinning of thick white which adversely affect the egg quality.

Montgomery[6] reported that at temperatures of 50 and 68°F, the loss of egg weight and quality were greater with warmer condition of storage

Yolk quality : The average initial yolk index of eggs was 0.45. In Fig. 2 under refrigerated storage upto 21 days the yolk index did not show rapid decline, it went down to 0.41 and 0.40 respectively on 10th and 21st days of storage at room temperature.

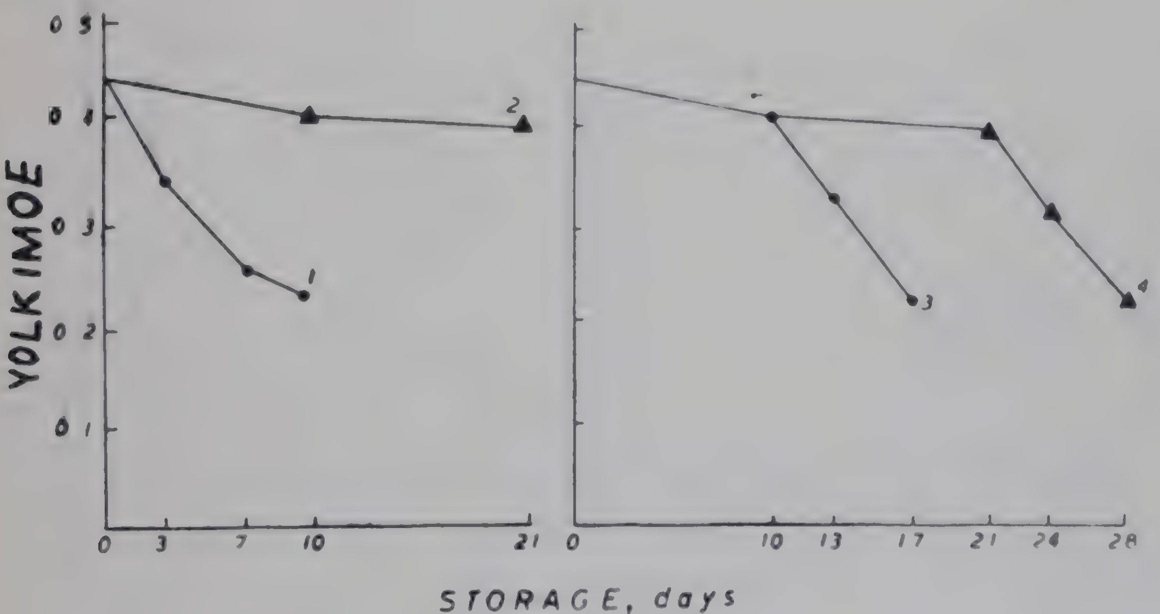

FIG. 2. Yolk index

1. Yolk index of egg at room temperature
2. Yolk index of egg at refrigerated temperature
3. Yolk index of egg 10 days in refrigerator and then keeping at room temperature for 3 and 7 days
4. Yolk index of egg 21 days in refrigerator and then keeping at room temperature 3 and 7 days

Eggs kept at room temperature showed considerable fall in yolk index from an initial of 0.45 to 0.35, 0.26 and 0.23, on 3rd and 7th and 10th day respectively.

In case of eggs kept at room temperature after refrigerated storage, the quality of eggs deteriorated considerably. The quality was evaluated on 3rd and 7th day at room temperature after taking out eggs on 10th and 21st days of refrigeration. The yolk index was found to be 0.33, 0.23, 0.31 and 0.22 respectively, the quality loss for later group was found to be more.

An initial yolk index of 0.42 and 0.40 and decline in the index to 0.38, 0.41 and 0.35 on the 7th and 21st days of refrigerated storage respectively have been reported by Khusu and Haleem[1]. They also found that yolk index declined to 0.33 and 0.34 from 0.42 and 0.4) on storage at air lock room on the 7th and 21st days respectively.

It is reported that[7] fresh egg quality in terms of Haugh unit, with the best quality at 79 Haugh units, and minimum at 60 Haugh units, and the yolk index for best quality at 0.45, and minimum at 0.39. In our study, initial yolk index was found to be 0.45 which is considered to be of best quality. It was found[7] that extremely poor quality eggs have a value around 0.22, while best quality eggs may have about 0.45. Eggs of a minimum acceptable quality should reach 0.39 on the scale. The size of the yolk in the newly laid eggs is a factor upon which the yolk index depends. The larger the yolk, the lower its index.

Haleem and Madhwaraj[8] reported initial yolk index of 0.45 from eggs collected from a commercial pullet flock. The yolk index declined to 0.40 and 0.11 res-

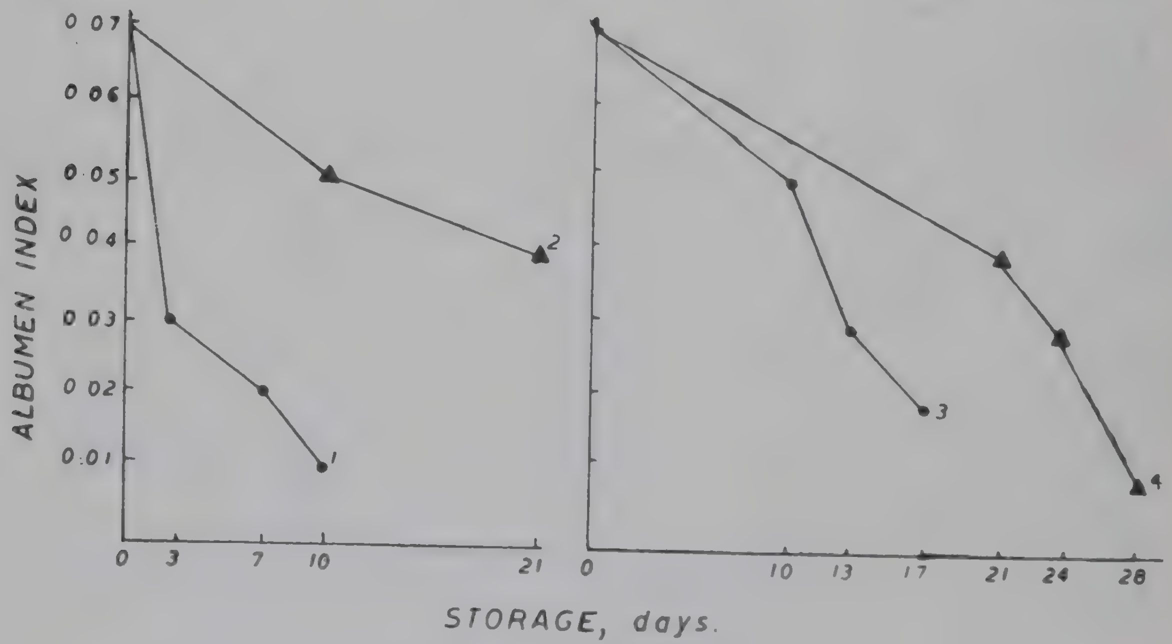

FIG. 3. Albumen index

1. Albumen index of egg at room temperature
2. Albumen index of egg at refrigerated temperature
3. Albumen index of egg 10 days in refrigerator and then keeping at room temperature for 3 and 7 days
4. Albumen index of egg 21 days in refrigerator and then keeping at room temperature for 3 and 7 days

pectively when the eggs were stored at 38°F., R. H., 80% and at 100°F, R. H. 72% (average) for unpacked stored eggs after 21 days of storage.

Albumen quality : The albumen index was 0.07 initially but dropped to 0.03, 0.02 and 0.01 on the 3rd, 7th and 10th day respectively when stored at room temperature (Fig. 3)

In the case of eggs stored in refrigerator and then at room temperature, the deterioration was rapid. In eggs stored 10 days in refrigerator and then at room temperature, for 3 and 7 days, the albumen index was 0.03 and 0.02 respectively. In the other batch of eggs stored at refrigerator temperature for 21 days and then at room temperature, the albumen index was found to be 0.03 and 0.01 respectively after 3 and 7 days.

In the case of refrigerated stored eggs, the quality declined slowly to 0.05 and 0.04, on 10th and 21st day but not so much as compared to other storage conditions.

Haleem *et al*[9] determined the albumen index of eggs of White Leg Horn birds for 3 consecutive periods of 16 weeks each. They found that initial albumen index of 0.10 declined to 0.09 and 0.07 at the end of 2nd and 3rd period of laying. In another study, Haleem and Madhwaraj[8] reported decline in albumen index from 0.07 to 0.04 and 0.01 respectively for storage conditions of 38°F and R.H. 80% and 100°F and R.H. 72%.

Khusu and Haleem[1] found initial albumen index of 0.04 and 0.06 from eggs in two age groups of same strain of layer. After 7 days of storage of the eggs in air lock room, the albumen became watery and its index could not be determined. In eggs stored in refrigerator for 7 and 21 days, the albumen index declined to 0.03 and 0.02 respectively.

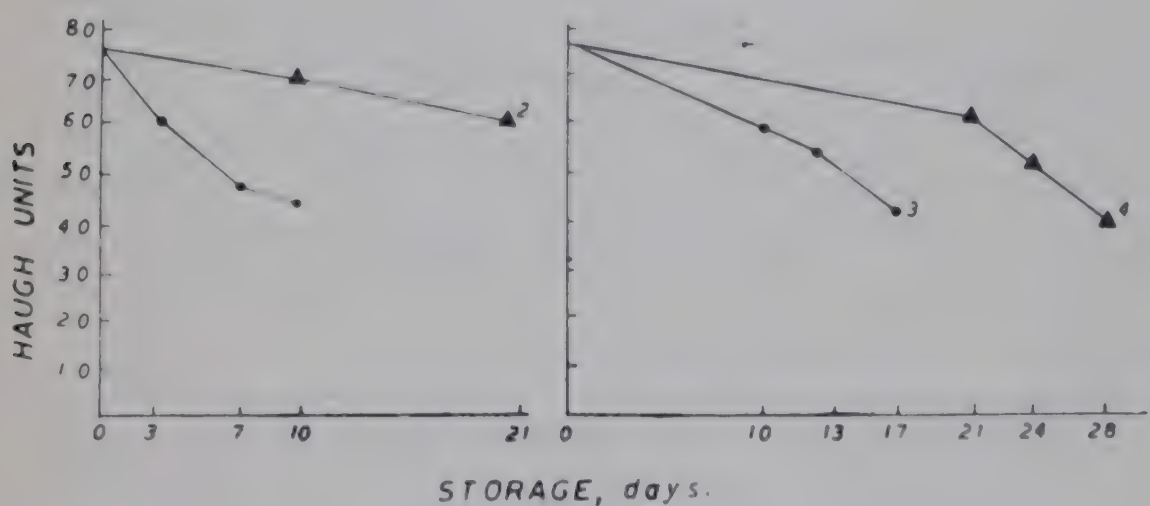

FIG. 4. Haugh units

1. Change in Haugh unit at room temperature
2. Change in Haugh unit at refrigerated temperature
3. Change in Haugh unit 10 days in refrigerator and then keeping at room temperature for 3 and 7 days
4. Change in Haugh unit 21 days in refrigerator and then keeping at room temperature for 3 and 7 days.

Wesley and Stdelman[10] concluded that per cent thick albumen is not as good indication of internal quality of eggs as some other measurements.

Haugh units: The average weight of the eggs was 54.56 g. The Haugh unit value of the eggs is presented in Fig. 4 which shows the initial Haugh units of 77 and its subsequent changes during storage. It showed slow changes in the eggs stored under refrigerated conditions. The initial Haugh unit value was 77 and on 10th day and 21st day, it was 71 and 62 respectively. At room temperature storage, it was found to be 62, 48 and 45 on the 3rd, 7th and 10th day.

In eggs stored for 10 days in refrigerator and then for 3 and 7 days in room temperature, Haugh units decreased to 55 and 42 respectively. In other batch, eggs stored at refrigerator temperature for 21 days and then for 3 and 7 days at room temperature, Haugh unit decreased to 51 and 40. Haleem *et al*[9] have reported a Haugh unit value of 90, 83 and 75 respectively for White Leg Horn layers for 1st, 2nd and 3rd each for 16 weeks period of laying. Haleem and Madhwaraj[4] have reported an initial Haugh unit of 82 for commercial pullet flocks, which declined to 59 and 60 after 21st day of refrigerated storage. The Haugh unit value was 24 at higher room temperature (100°F, R. H. 72%) after 2 weeks of storage.

Khusu and Haleem[1] also observed slow changes in Haugh unit value for the eggs stored under refrigerated conditions. The initial Haugh unit value of commercial layer eggs was 59 and 64 for eggs collected on 9th and 12th months of laying periods. The Haugh unit value was better for initial and 7 days refrigerated storage, although, it was not so for other storage periods when compared to eggs collected on 9th month of laying period.

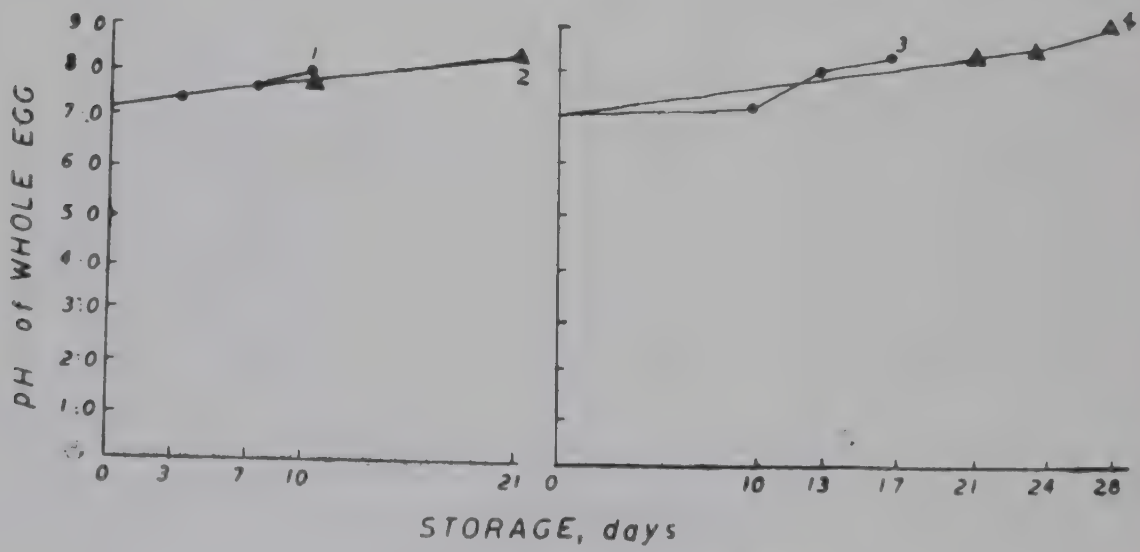

FIG. 5. pH Value of whole eggs

1. pH of whole egg at room temperature
2. pH of whole egg at refrigerated temperature
3. pH of whole egg 10 days in refrigerator and then keeping at room temperature for 3 and 7 days
4. pH of whole egg 21 days in refrigerator and then keeping at room temperature for 3 and 7 days.

pH : The initial pH of the whole egg was 7.12 and after storage at room temperature for 3, 7 and 10 days, it increased to 7.23, 7.40 and 7.85 respectively (Fig. 5).

In the case of refrigerated temperature, the pH was 7.5 on the 10th and 8.47 on the 21st day.

The eggs kept for first 10 days in refrigerator and then 3 and 7 days in room temperature, showed a pH of 8.20 and 8.52 respectively. In the other batch where eggs were kept for 21 days in refrigerated storage and then 3 and 7 days in room temperature, the pH was found to be 8.52 and 9.10 respectively.

Summary

Loss in weight of eggs under refrigerated storage was low, when compared to storage at room temperature but in the case of refrigerated storage followed by room temperature storage, the weight losses were quite high. The loss in weight of eggs kept at room temperature was highest. The yolk index showed a little variation in refrigerated storage compared to marked difference in room temperature storage. The albumen index, Haugh units, yolk index and pH declined considerably when refrigerated storage of eggs was followed by room temperature storage.

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SESSION II

Heat Transfer and Thermodynamics of Refrigeration

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Heat Pipe Technique of Heat Transfer in Cryogenic Fields

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Introduction

A 'Heat Pipe' is a relatively recent technique used in transferring heat from a hot source to cooler source, far more rapidly and more conveniently than any method known so far. The fluid used in this technique picks up heat from the hot source, gets itself vaporised, and by virtue of its pressure, travels quickly to the cooler source some distance away. There the vapour gets itself condensed to a liquid, thereby transferring all its latent heat to the cooler source and the condensate gets itself pumped back to the hot source by capillary action, and the same cycle continues. There are no moving parts and no external power is required for the circulation of the heat transfer fluid in the heat pipe. There are numerous applications of the 'Heat Pipe', wherein heat from a hot source awkwardly located could be made to transfer heat very quickly to any convenient location. The paper discusses the principle, construction, and mathematical function of the pipe, and deals with its useful application in refrigeration, airconditioning and cryogenic fields.

The novel idea of the heat pipe was suggested as early as in 1942, by Gaugler. However, no interest was shown in its development and the technique remained dormant for another twenty years. Later in 1963, Grover[1] took a patent for this development and perfected it. That heat pipe, thereafter, became a centre of great attraction and within the last ten years there has been rapid advancement towards its perfection. The recent energy crisis has also helped in making it a valuable tool to utilise waste heat in various industries.

Construction of the Heat Pipe

The general construction and operation of the heat pipe is shown in Fig. 1.

The heat pipe can be divided into three parts: 1. The working fluid, 2. The wick, and 3. The container.

The selection of working fluid depends on the operating temperature ranges. Use of fluorocarbons, freons, methane, liquid oxygen, liquid nitrogen is common in refrigeration and cryogenic heat transfer applications. Various liquid metals (sodium, potassium) are used in high temperature applications. Water is the most common working fluid used in normal temperature ranges. Fluid properties such as (1) heat of vaporisation, (2) capillary action, (3) viscosity and (4) vapour pressure at operating temperature range dictate the selection of the working fluid.

Wick of the heat pipe acts as an effective capillary pump. A wide variety of wicks are used in heat pipes. The most widely used one is the wick made of several

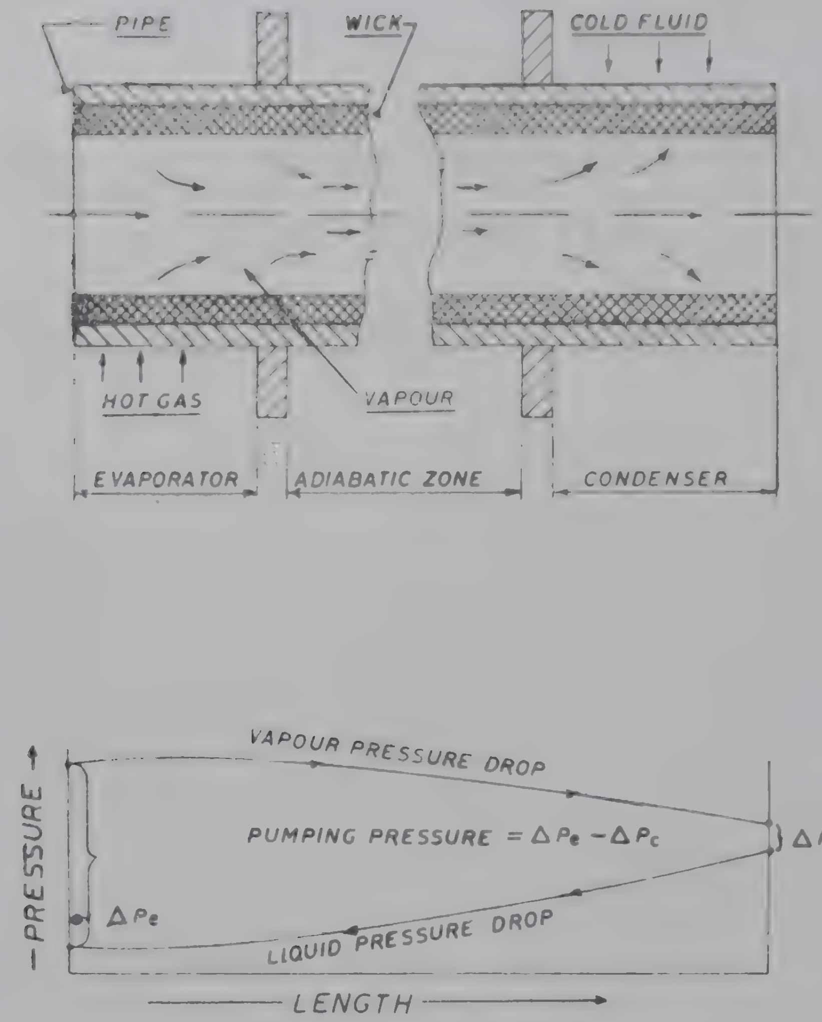

FIG. 1. Heat pipe operational principle

layers of fine mesh screen. Several textile fabrics, porous metals, ceramic tubes, sintered copper fibre wick, channels cut into the interior surface running axially along the length of the tube can also be used as wicks.

The container of the heat pipe is a metal tube and it should have high thermal conductivity. It must withstand the pressure difference between the inside and the outside of the heat pipe and high internal pressure due to the vapour pressure of the working fluid.

Axially it can be divided into three parts. The evaporator, adiabatic zone and the condenser. Evaporator is the part of the heat pipe which comes in contact with the heat source and the part which comes in contact with the heat sink is referred to as the condenser. Many a times, adiabatic region of the heat pipe is compatible to any geometrical shapes. A variety of cross sectional shapes are used in heat pipe construction. Circular cross-section is the most common. It can be rigid or flexible.

Function of the Heat Pipe

Thermal energy is transferred from the evaporator to the condenser by evapo-

ration and condensation of the working fluid. The condensate is returned to the evaporator by the surface tension forces developed in the wick. The wick is lined inside the container and is saturated with the working fluid. Thermal energy from the source to the working fluid is transferred by conduction through the container and the wick. This results in the evaporation of the working fluid. A vapour-liquid interface is formed. As more and more heat is transferred, more and more mass is evaporated and then the vapour-liquid interface recedes into the wick and the resulting concave shape of the interface is responsible for the working of the heat pipe. Vapour flows to the condenser end due to the small pressure difference in the vapour core [This is due to slight temperature difference in the evaporator and the condenser]. This temperature drop is very small (of the order of 1 to 1.5°C) and thus heat pipe could be considered as operating isothermally. As vapour flows, its pressure drops due to the viscosity effects. Vapour condenses in the condenser and vapour-liquid interface becomes flat i.e. radius of curvature of interface tending to infinity. The heat of condensation flows to the heat sink by conduction. Condensate is pumped through the wick to the evaporator by capillary action.

Mathematical Model

Let

p_L = Pressure drop in porous wick, due to condensate flow [kgf/m²]

p_v = Pressure drop in heat pipe, due to vapour flow [kgf/m²]

p_g = Pressure lost due to gravity force acting on the condensate [kgf/m²]

p_g' = Pressure lost due to gravity force acting on vapour [kgf/m²]

p_t = Pumping pressure available due to surface tension force in the wick [kgf/m²]

μ_L = Viscosity of condensate $\left[\frac{\text{kgf} - \text{sec}}{\text{m}^2}\right]$

μ_v = Viscosity of vapour $\left[\frac{\text{kgf} - \text{sec}}{\text{m}^2}\right]$

l = Effective length of heat pipe (meters) i.e. from centre of evaporator to centre of condenser

$\dot{M}$ = Mass flow rate (kg/hr)

ρ_L = Density of condensate (kg/m³)

ρ_v = Density of vapour (kg/m³)

a = Total cross-sectional area of wick (m²)

χ = Porosity factor of material used for condensate flow

K = Wick permeability (m²)

δ = Flow factor

A = Cross-section of vapour flow (m²)

r_v = Radius of passage, through which vapour flows (meter)

r_c = Hydraulic radius (meter) of the wick pore at the liquid vapour interface, on the condenser side

$$\left[\text{hydraulic radius} = \frac{2\,(\text{pore area})}{\text{wetted perimeter}} \right]$$

r_e = Hydraulic radius (meter) of the wick pore at the liquid vapour interface, on evaporator side

ϕ_c = Liquid contact angle, on condenser side °

ϕ_e = Liquid contact angle, on evaporator side °

σ = Liquid surface tension (kgf/meter)

θ = Inclination of heat pipe with vertical °

L = Latent heat of liquid at average saturation temp. (Kcal/kg)

Q = Rate at which heat is transferred through heat pipe, from the evaporator to the condenser (Kcal/hr)

Cotter[2] has presented first mathematical model of the heat pipe.

The heat pipe is so designed that sufficient driving force is made available, due to surface tension of the wick, to overcome the consequent resistance in the system.

The basic equation for the design is then,

Available driving force $\geq$ Resistance of system.

$$\therefore\ p_t \geq p_L + p_v \pm p_g \pm p_g'$$

Now it may be shown that

$$p_t = 2\sigma \left[\frac{\cos\phi_e}{r_e} - \frac{\cos\phi_c}{r_c} \right] \quad \ldots\ldots \quad (1)$$

$$p_L = \frac{\mu_L \cdot l \cdot \dot{M}}{\rho_L\ K.a_L} \quad \ldots\ldots \quad (2)$$

Darcy formula

$$p_v = \frac{8\mu_v\ \dot{M}.\delta \cdot l}{(r_v)^2 \rho_v A} \quad \ldots\ldots \quad (3)$$

Where the factor δ depends on whether the vapour flow is laminar or turbulent.

$$p_g = \rho_L\ .l.\cos\theta.g \quad \ldots\ldots \quad (4)$$

$$p_g' = \rho_v.l.\cos\theta.g \quad \ldots\ldots \quad (5)$$

For gravitational pressures p_g and p_g', plus sign is to be used for p_g and minus for p_g' when condensate flows downwards, i.e. in direction of gravity. This will happen when the condenser side of heat pipe is above the evaporator side.

Similarly minus sign for p_g and plus sign for p_g' be used when condenser is below and evaporator is above.

It will be noted that p_g' will be too small a quantity compared to p_g and so we neglect the term.

Substituting and simplifying the basic equation and knowing that

$$Q = \dot{M}.L.$$

We have the final design equation as

$$Q = \frac{L.2\sigma \left[\dfrac{\cos\phi_e}{r_e} - \dfrac{\cos\phi_c}{r_c} \right] - \rho_L\ .g.\cos\theta.l.L}{\dfrac{\mu_L \cdot l}{\rho_L\ K.a} + \dfrac{\delta.8\mu_v.l}{\rho_v.A.(r_v)^2}}$$

Limitations of Heat Pipe

The function of heat transfer by use of heat pipe is likely to be affected due to the following phenomena :

(*a*) "Wicking Limit" i.e. If the heat input to evaporator is faster than the heat taken out from condenser side of the tube, the vapour would travel faster than the speed at which the condensate returns through the wick, This results in drying of wick and ultimately it may turn out.

(*b*) "Sonic Limit" i.e. If in a given design of heat pipe, for a given fluid to be heated, the incoming fluid temperature happens to be too low the high rate of condensation causes drop of pressure and this results in a high rush of vapour from the evaporator side. If such increase in velocity reaches sonic value, the flow of vapour gets choked, and the Heat Pipe function ceases.

(*c*) "Entrainment Limit" i.e. A faster rate of evaporation causes entrainment of liquid particles with the vapour, which also causes dry out of the wick [Fig. 2 indicates the limit of heat flux transferred in a heat pipe, as imposed by the above three limiting factors].

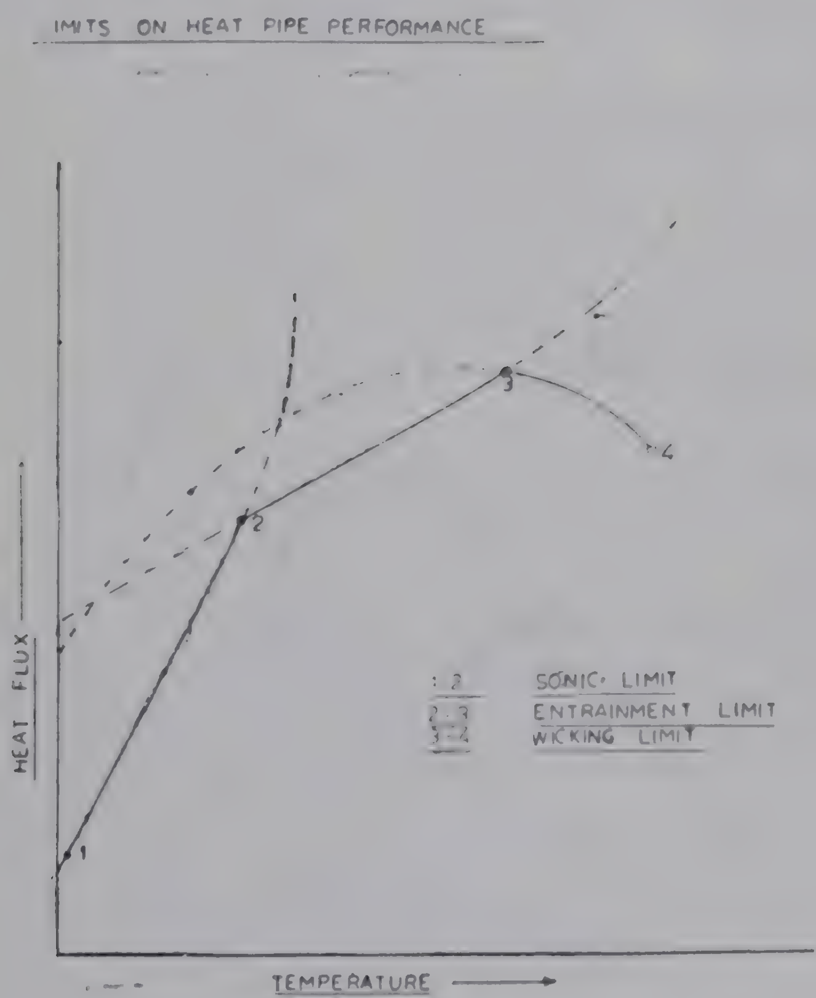

FIG. 2. Limits on heat pipe performance

Application of the Heat Pipe

Heat pipe finds its applications due mainly to the following features :

1. Nearly isothermal and high rate of heat transfer.
2. No requirement of external power nor gravitational force.

Some possible fields of application of heat pipes in Refrigeration, Airconditioning, and cryogenics are listed below :

(*a*) Mackinney[3] recommended the installation of heat pipe around a cryogenic storage tank. If heat pipe contains a working fluid having lower boiling point than the stored fluid, it should be feasible to transfer the heat leaked from the surrounding area into the storage vessel. Heat pipe mounted on storage tanks of cryogenic fluids used as propellants will be ideal for space applications.

(*b*) Katzoff[4] recommended the use of long heat pipes wrapped around the circumference of space craft to equilize the temperature distribution. Evaporator of heat pipe faces the sun (Fig. 3) and radiation cooled condenser remains in shadow of the craft. Such arrangement would reduce the temperature variations around a space craft. NASA has successfully employed such heat pipe on 'Appolo' space craft.

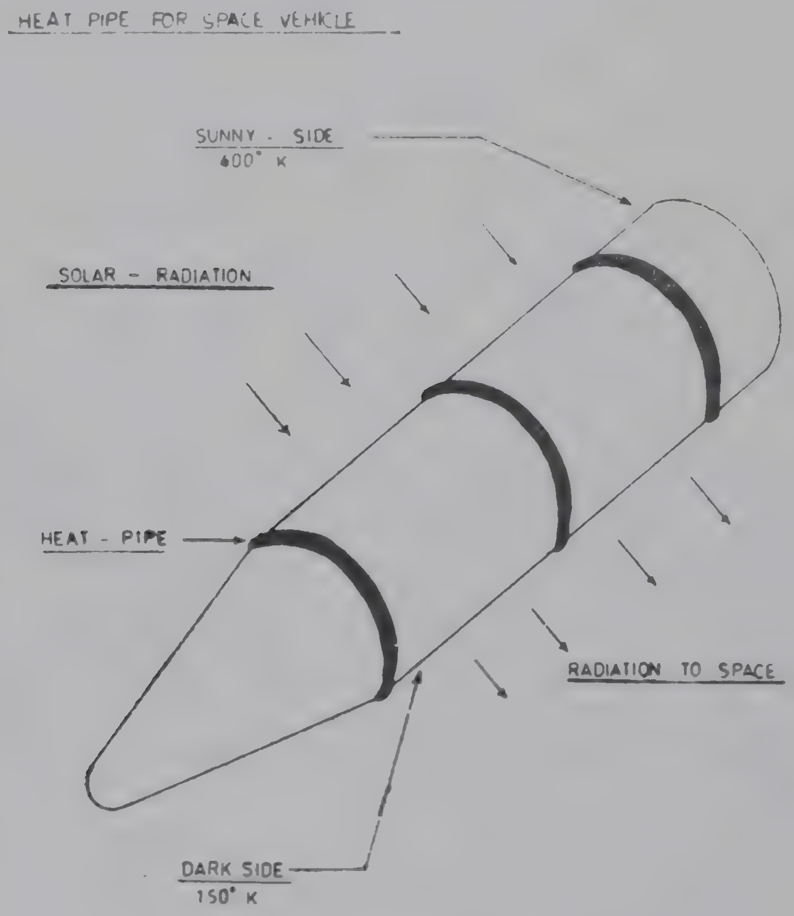

Fig. 3. Heat pipe for space vehicle

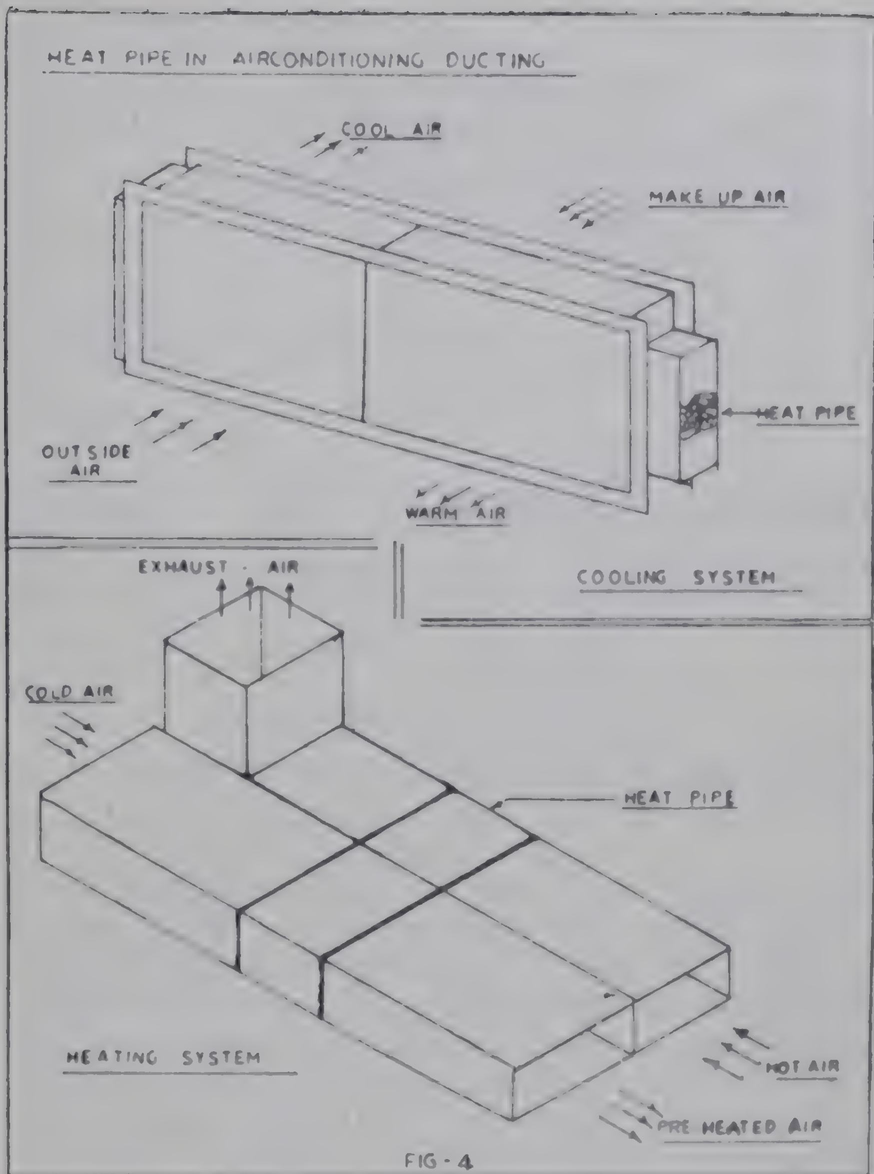

FIG. 4. Heat pipe in airconditioning ducting

(*c*) Donald[5] has suggested use of heat pipe in air conditioning ducting system to transfer thermal energy using a refrigerant as working fluid. Such application is shown in Fig. 4. Due to high heat transfer coefficients, heat pipe as a heat exchanger is very compact compared to conventional heat exchangers. It is reported that the heat pipe could recover 58% to 78% thermal energy from the exhaust air and adds to make up air.

(*d*) Heat pipe has also been used for cryogenic cooling infra red detectors[6]. On satellites the NASA ICICLE system proposes to use several cryogenic temperature heat pipes to thermally connect a refrigerator to the detectors.

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Computer Programme for Calculating Thermodynamic Properties of R-22 in S. I. Units

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Introduction

With the adoption of S. I. units it is necessary that the thermodynamic properties of various refrigerants are made available in these units. For this purpose it is more convenient to calculate these properties *ab initio* from the basic equations rather than obtain them by conversion of units from existing tables. A computer programme for calculating thermodynamic properties has also the advantage that it can be used for computerized analysis of refrigeration cycles and systems. The purpose of this paper is to present complete computer programmes for the calculation of the thermodynamic properties of R-22 in S. I. units.

Method of Computation

The method of computing thermodynamic properties of refrigerants is well documented[1,2]. For this reason only a brief outline is given below. Four basic equations are usually employed. For R-22 these equations have the form given below :

Equation of State :

$$P = \frac{RT}{V-B} + \frac{A_2 + B_2 T + {}_2C\, e^{-kT/T_c}}{(V-B)^2} + \frac{A_3 + B_3 T + C_3\, e^{-kT/T_c}}{(V-B)^3} + \frac{A_4 + B_4 T + C_4\, e^{-kT/T_c}}{(V-B)^4} + \frac{A_5 + B_5 T + C_5\, e^{-kT/T_c}}{(V-B)^5} + (A_6 + B_6 T)\, e^{-\alpha V} \quad \ldots\ldots (1)$$

Vapour Pressure :

$$
\begin{aligned}
Log_{10} P = AVP + \frac{BVP}{T} &+ CVP \log_{10} T \\
&+ DVP \cdot T + \frac{EVP(FVP - T)}{T} \log_{10}(FVP - T) \\
&+ CVP\left(\frac{FVP - T}{T}\right) \qquad \ldots\ldots \quad (2)
\end{aligned}
$$

Constant Volume Specific Heat of Vapour at Zero Pressure :

$$
C_v = ACV + BCV \cdot T + CCV \cdot T^2 + \frac{FCV}{T^2} \qquad \ldots\ldots \quad (3)
$$

Liquid Density :

$$
\begin{aligned}
D_l = AL + BL\left(1 - \frac{T}{T_c}\right)^{1/3} &+ CL\left(1 - \frac{T}{T_c}\right)^{2/3} + DL\left(1 - \frac{T}{T_c}\right) \\
&+ EL\left(1 - \frac{T}{T_c}\right)^{4/3} \qquad \ldots\ldots \quad (4)
\end{aligned}
$$

The values of the constants in the above equations and the critical temperature T_c for R–22 are listed below. The units are : pressure = Newton/cm², temperature = °K, volume = m³/kg, specific heat = kJ/kg — °K.

R	=	0.0096146747	T_c	=	369.18
B	=	$0.1248556356 \times 10^{-3}$	α	=	8781.3417
A_2	=	—0.0116981906	k	=	4.2
B_2	=	$0.116431239 \times 10^{-4}$	AVP	=	27.18937909
C_2	=	—0.1184097983	BVP	=	−2136.218417
A_3	=	$-0.29295248 \times 10^{-5}$	CVP	=	−7.86103122
B_3	=	0.2303194×10^{-7}	DVP	=	0.0039435102
C_3	=	0.248896×10^{-8}	EVP	=	0.445746703
A_4	=	0.241918×10^{-7}	FVP	=	381.167
B_4	=	$-0.6796653 \times 10^{-10}$	GVP	=	0.1137857608
C_4	=	0.0	ACV	=	0.1177678176
			BCV	=	$0.1699729598 \times 10^{-2}$
A_5	=	$-0.2434407562 \times 10^{-10}$	CCV	=	$-0.8830432915 \times 10^{-6}$
B_5	=	$0.6301775665 \times 10^{-13}$	FCV	=	332.5417586
C_5	=	$-0.1206109838 \times 10^{-9}$	AL	=	524.76606
			BL	=	875.1612853
A_6	=	$0.9400226152 \times 10^{8}$	CL	=	588.662575
B_6	=	$-0.2075806496 \times 10^{6}$	DL	=	−357.0934636
			EL	=	327.9513737

It is conventional to assign zero enthalpy and zero entropy to saturated liquid at—40°C(233.15°K). With this datum, the procedure for calculating the various thermodynamic properties is as follows.

Saturation Pressure and Temperature

The saturation pressure corresponding to a given temperature is directly obtained from Eq. (2). In order to determine the saturation temperature corresponding to a given pressure P, an iterative procedure is used. The first estimate T_1 of saturation temperature is computed from the approximate relation

$$T_1 = 101.5 + 215.5 \log_{10} P \qquad \ldots\ldots \qquad (5)$$

The Newton-Raphson method is then used to find the correct value of saturation temperature as follows. Let

$$F = AVP + \frac{BVP}{T_1} + CVP\, log_{10}\, T_1 + DVP \,.\, T_1 + \frac{EVP\,(FVP - T_1)}{T_1}\, log_{10}\,(FVP - T_1) + \frac{GVP\,(FVP - T_1)}{T_1} - log_{10}\, P \qquad \ldots\ldots \qquad (6)$$

$$FP = \frac{dF}{dT} = -\frac{BVP}{T_1^2} + \frac{CVP}{C_1 . T_1} + DVP - \frac{EVP}{C_1 . T_1} - \frac{EVP\,.\,FVP}{T_1^2\ .\ C}\, log_e\,(FVP - T) - \frac{GVP\ ,\ FVP}{T_1^2} \qquad (7)$$

where $C_1 = 2.302585$. The second approximation of T_{sat} is then given by

$$T_2 = T_1 - \frac{F}{FP} \qquad \ldots\ldots \qquad (8)$$

The iteration is continued until two successive values of T_{sat} do not differ by more than 0.01°C.

Specific Volume of Saturated Liquid, V_f

At any given temperature the saturated liquid density D_f is directly calculated from Eq. (4). V_f is then given by

$$V_f = 1/D_f \qquad \ldots\ldots \qquad (9)$$

Specific Volume of Saturated Vapour V_g or of Superheated Vapour V

At a given pressure P and temperature T (or P_{sat} and T_{sat}) the specific volume V (or V_g) is obtained by iteration from Eq. (1) using the Newton-Raphson method. For this purpose the first estimate V_1 of V is obtained by assuming ideal gas relationship

$$V_1 = \frac{R\,T}{P} \qquad \ldots\ldots \qquad (10)$$

Enthalpy of Vaporisation H_{fg} and Entropy of Vaporisation S_{fg}

These are determined at a given temperature T by using the Clapeyron equation and the vapour pressure relation[2]. From Clapeyron equation

$$H_{fg} = T\,(V_g - V_f)\,\frac{d\,P}{d\,T} \qquad \ldots\ldots \qquad (11)$$

By using Eq. (2) we get

$$H_{fg} = 10 \times P_{sat} \times (V_g - V_f)\Big[-\frac{BVP \cdot C_1}{T} + CVP + DVP \cdot C_1 \cdot T$$

$$- EVP - \frac{EVP \cdot FVP}{T} \log_e (FVP - T)$$

$$- \frac{FVP \cdot GVP \cdot C_1}{T}\Big] \qquad \ldots\ldots \quad (12)$$

where $C_1 = 2.302585$ and the constant 10 has been used to give H_{fg} in kJ/kg. The entropy of vaporisation is simply given by

$$S_{fg} = \frac{H_{fg}}{T} \qquad \ldots\ldots \quad (13)$$

Enthalpy of Saturated Vapour H_g or of Superheated Vapour H

Suppose we have to determine H (or H_g) at state T, P (or T_{sat}, P_{sat}). The specific volume at this state, V, is obtained from Eq. (1) as explained previously. In order to compute the enthalpy at state T, P, V, let us reach this state from the datum state by following the path 1-2-3-4-5 as shown in Fig. 1. States 3 and 4 are ideal gas states at ($P \to 0$, $V \to \alpha$) at temperature T_o (−40°C) and T, respectively. Then

$$H_5 = H_1 + (H_2 - H_1) + (H_3 - H_2) + (H_4 - H_3) + (H_5 - H_4)$$

$$= O + (H_{fg})_{T_o} + \Big(-\int_{\alpha}^{V_2} [T_o \left(\frac{\delta P}{\delta T}\right)_V - P] \, dV + RT_o - P_2 V_2\Big)$$

$$+ \int_{T_o}^{T} (C_V + R) \, dT + \Big(\int_{\alpha}^{V_5 = V} [T\left(\frac{\delta P}{\delta T}\right)_V - P] \, dV + PV - RT \qquad \ldots \quad (14)$$

Note that $P_5 = P$, $V_5 = V$, $T_5 = T$, and that we have used the relation

$$\left(H - H_{ideal}\right)_{P,V,T} = \int_{\alpha}^{V} [T\left(\frac{\delta P}{\delta T}\right)_V - P] \, dV + PV - RT \qquad \ldots \quad (15)$$

By using Eq. (1) in Eq. (14) :

$$H = [(H_{fg})_{T_o}] - [10.P_2 \; V_2] - 10 \cdot \Big[\frac{A_2}{V_2 - B} + \frac{A_3}{2(V_2 - B)^2}$$

$$+ \frac{A_4}{3(V_2 - B)^3} + \frac{A_5}{4(V_2 - B)^4} + \frac{A_6}{\alpha} e^{-\alpha V_2} + \left(1 + \frac{k \; T_o}{T_c}\right) e^{-kT/T_c}$$

$$\Big\{ \frac{C_2}{V_2 - B} + \frac{C_3}{2(V_2 - B)^2} + \frac{C_4}{3(V_2 - B)^3} + \frac{C_5}{4(V_2 - B)^4} \Big\} \Big]$$

$$- [ACV \cdot T_o + BCV \cdot \frac{T_o^2}{2} + CCV \cdot \frac{T_o^3}{3} - \frac{FCV}{T_o}]$$

$$+ (ACV \,.\, T + BCV \,.\, \frac{T^2}{2} + CCV \,.\, \frac{T^3}{3} - \frac{FCV}{T}) + 10 \,.\, PV$$

$$+ 10 \,.\, [\frac{A_2}{V-B} + \frac{A_3}{2(V-B)^2} + \frac{A_4}{3(V-B)^3} + \frac{A_5}{4(V-B)^4} + \frac{A_6}{\alpha} e^{-\alpha V}$$

$$+ (1 + \frac{k\ T}{T_c}) e^{-kT/T_c} \left\{ \frac{C_2}{V-B} + \frac{C_3}{2(V-B)^2} + \frac{C_4}{3(V-B)^3} + \frac{C_5}{4(V-B)^4} \right\}] \quad \ldots (16)$$

The terms in the first four square brackets on the RHS of Eq. (16) are all constant and as such need be computed only once. The value of this constant for R-22 is 145.14681 kJ/kg.

Entropy of Saturated Vapour Sg or of Superheated Vapour S

Referring to Fig. 1, the entropy at state 5 (P, V, T) is given by

$$S_5 = S_1 + (S_2 - S_1) + (S_3 - S_2) + (S_4 - S_3) + (S_5 - S_4)$$

$$= 0 + (S_{fg})_{T_o} + \underset{T=T_o}{\int_V^{\infty}} \left(\frac{\delta P}{\delta T}\right)_V dV + \int_{T_o}^{T} \frac{CV}{T} dT + \underset{T=T}{\int_{\infty}^{V}} \left(\frac{\delta P}{\delta T}\right)_V d_V \quad \ldots\ldots (17)$$

Using Eq. (1) with Eq. (17) we get :

$$S = [S_{fg}]_{T_o} + 10 \,.\, \left[\frac{B_2}{V_2-B} + \frac{B_3}{2(V_2-B)^2} + \frac{B_4}{3(V_2-B)^3} + \frac{B_5}{4(V_2-B)^4} + \frac{B_6 e^{-\alpha V_2}}{\alpha} \right]$$

$$- 10 \,.\, \left[\left(\frac{k}{T_c} e^{-kT_o/T_c} \right) \left\{ \frac{C_2}{V_2-B} + \frac{C_3}{2(V_3-B)^2} + \frac{C_4}{3(V_2-B)^3} + \frac{C_5}{4(V_2-B)^4} \right\} \right]$$

$$- \left[ACV \,.\, \log_e T_o + BCV \,.\, T_c + \frac{CCV \cdot T_o^2}{2} - \frac{FCV}{2\,T_o^2} \right]$$

$$+ 10 \,.\, R \log_e \left(\frac{V-B}{V_2-B} \right) - 10 \,.\, \left[\frac{B_2}{V-B} + \frac{B_3}{2(V-B)^2} + \frac{B_4}{3(V-B)^3} \right.$$

$$\left. + \frac{B_5}{4(V-B)^4} + \frac{B_6 e^{-\alpha V}}{\alpha} \right]$$

$$+ 10 \,.\, \left(\frac{k}{T_c} e^{-kT/T_c} \right) \left[\frac{C_2}{V-B} + \frac{C_3}{2(V-B)^2} + \frac{C_4}{3(V-B)^3} + \frac{C_5}{4(V-B)^4} \right] \quad \ldots (18)$$

The first four square brackets on the RHS of Eq. (18) can be combined into a single constant. The value of this constant for R - 22 is -0.00462254 kJ/kg — °K.

Enthalpy and Entropy of Saturated Liquid, H_f and S_f

These are easily obtained from :

$$H_f = H_g - H_{fg} \quad \ldots\ldots (19)$$

$$S_f = S_g - S_{fg} \quad \ldots\ldots (20)$$

Computer Programmes

Four computer programmes are presented. The first two programmes are to evaluate respectively the saturation temperature at a given pressure, and the specific volume of vapour at given pressure and temperature. The remaining two programmes are for the computation of the thermodynamic properties of saturated vapour and superheated vapour. The results obtained from these programmes are in perfect agreement with the values given in the ASHRAE tables.

```
C  C    Function Prog.  Tsat (Psat)  F—22
        Read 111, AVP, BVP, CVP, DVP, EVP, FVP, GVP, CL
111     Format (4 E 20.10/4 E 20.10)
99      Read, Psat
C       Initial Estimate of Tsat
113     Plog=Log (Psat)/C1
        TK=101.87+215.5* Plog
        Iter=0
C       Iterate using Newton iteration
11      Iter=Iter+1
        TKO=TK
        IF (iter—30) 2,2,5
2       C=log (ABS (FVP—TKO))/C1
        BB=GVP*  (FVP—TKO)/TKO—PLOG
115     F=AVP+BVP/TKO+CVP* log (TKO)/C1+DVP*TKO+EVP*(FVP—TKO)*C/TKO+BB
        FP=—BVP/(TKO* TKO)+CVP/(C1* TKO)+DVP—EVP/(C1* TKO)
        FP=FP—GVP* FVP/(TKO* TKO)—EVP* C* FVP/(TKO* TKO)
120     TK=TKO—F/FP
        IF(ABS(TK—TKO)—0.01) 5,5,11
5       Tsat=TK
        Punch 89, Psat, Tsat
89      Format  (2f 20.8)
        Go to 99
        Stop
        End
   27.18937909          —2136.218417          —7.86103122          0.0039435102
    0.445746703           381.167               0.1137857608         2.302585093
   10.4952
   161.8269
C  C    Function Prog.  Tsat (Tsat)
          10.49520000          233.16255000
         161.82690000          315.38634000
C  C    Specific volume SVOLF (TK, PN)
        Equation of state constants
101     Format (4 E 20.10/4 E 20.10/4 E 20.10/4 E 20.10/3 E 20.10)
        Read 101, R, B, A2, B2, C2, A3, B3, C3, A4, B4, C4, A5, B5, C5, XK, TCRIT, A6, B6,
        1alpha
3       Read, tk, pn
        T=TK
        E=2.71828** (—XK* T/TCRIT)
        E1=PN
        E2=R* T
        E3=A2+B2* T+C2, E
```

```
          E4=A3+B3* T+C3*E
          E5=A4+B4* T+C4* E
          E6=A5+B5* T+C5* E
          E8=0.0001
          E32=2. *E3
          E43=3. *E4
          E54=4. *E5
          E65=5. *E6
          E7=A6+B6*
C         Compute initial estimate of V from ideal gas law
          VN=R* T/PN
          Iter—0
C         Compute V by Newton iteration
1         Iter=Iter+1
          IF (Iter—30) 2, 2, 20
2         V=VN
          V2=V*V
          V3=V2* V
          V4=V3* V
          V5=V4* V
          V6=V5* V
          EMAV=—alpha*(V+B)/2070
          EMAV=2.71821**EMAV
          EMAV=EMAV**46
          EMAV=EMAV**45.0
          F=E1—E2/V—E3/V2—E4/V3—E5/V4—E6/V5—E7* EMAV
          FV=F2/V2+E32/V3+E43/V4+E54/V5+E65/V6+E7* ALPHA* EMAV
          VN=V—F/FV
          IF (ABS ((VN—V)/V)—E8) 20, 20, 1
20        SVOLF=VN+B
          Punch, TK, PN, SVOLF
          Go to 3
          Stop
          End
    0.9614670OE—02       0.1248563600E—03    —0.1169819000E—01     0 1164312400E—04
   -0.1184098000        —0.2929524800E—05     0.2303194000E—07     0.2488960000E—03
    0.2419180000E—07   —0.6796653000E—10      0.0000000000        —0.2434407600E—10
    0.6301775700E—13   —0.1206109800E—09      0.4200000000E - 01    0.3691800000E+03
    0.9400226200E+08   —0.2075806500E+06      0.8781341700E+04
  233.16,    10.495
  315.38,    161.82
C C       Saturation properties F - 22
C         Equation of state constants
101       Format (4E20.10/4E20.10/4E20.10/4E20.10/3E20.10)
          Read 101, R. B. A2, B2, C2, A3.B3, C3, A4, B4, C4, A5, B5, C5, XK, TCRIT, A6, B6.
          Ialpha
C         Liquid density constants
          Read 102, AL, BL, CL, DL, EL
102       Format (5F15,8)
C         Specific heat constants
          Read 111, AVP, BVP, CVP, DVP, EVP, FVP, GVP
111       Format (4F 20.10/3F 20.10)
          Read, ACV, BCV, CCV, FCV
```

```
      C1=2.302585093
      XH=34.66772*4.1868
      XS=—0.00462254
5     Read 6, TK
6     Format (F 20.8)
      T=TK
C     Calculate Psat
      C=log (FVP—T)/C1
      Psat1=AVP + BVP/T + CVP* LOG (T)/C1 + DVP* T + EVP* (FVP — T)* C/T + GVP*
        (FVP—T)/T
      Psat—10.** Psat1
C     Calculate VG
      PN=Psat
      T=TK
      E=2.71828** (—XK* T/TCRIT)
      E1=PN
      E2=R* T
      E3=A2+B2* T+C2* E
      E4=A3+B3* T+C3* E
      E5=A4+B4* T+C4* E
      E6=A5+B5* T+C5* E
      E8=0.0001
      E32=2.* E3
      E43=3.* E4
      E54=4.* E5
      E65=5.* E6
      E7=A6+B6* T
C     Compute initial estimate of V from ideal gas law
      VN=R* T/PN
      Iter=0
C     Compute V by Newton iteration
1     Iter=iter+1
      IF (iter—30) 2, 2, 20
2     V=VN
      V2=V* V
      V3=V2* V
      V4=V3* V
      V5=V4* V
      V6=V5* V
      EMAV=—alpha* (V+B)/2070.
      EMAV=2.71828** EMAV
      EMAV=EMAV**46.
      EMAV=EMAV**45.0
      F=E1—E2/V—E3/V2—E4/V3—E5/V4—E6/V5—E7* EMAV
      FV=E2/V2+E32/V3+E43/V4+E54/V5+E65/V6+E7* alpha* EMAV
      VN=V—F/FV
      IF (ABS ((VN—V)/V)—E8) 20, 20, 1
20    SVOLF=VN+B
      VG=SVOLF
C     Calculate VF
      TK1=1.0—T/TCRIT
      TK13=TK1**(1./3.)
      TK23=TK1**(2./3.)
      TK43=TK1**(4./3.)
```

```
          DF=AL+BL* TK13+CL* TK23+DL* TK1+EL* TK43
55        VF=1./DF
C         Calculate HFG by Clausius Clapeyron eqn.
56        BKT=—BVP* C1/T+CVP+DVP* C1* T—EVP—EVP* FVP* log (FVP—T)/T—FVP*
            GVP* C1/T
          HFG=(VG—VF)* Psat* BKT* 10.
          SFG=HFG/T
C         Calculate HG and SG
58        T2=T* T
          T3=T2* T
          T4=T3* T
          V1=VG—B
59        V2=2.* V1* V1
          V3=3.* V1** 3
          V4=4.* V1** 4
          XKT=XK* T/TCRIT
          XKT1=2.71**(—XKT)
61        EMAV=—alpha* VG/2070.
          EMAV=2.71828** EMAV
          EMAV=EMAV** 46
          EMAV=EMAV** 45
64        H1=ACV* T+BCV*T2/2.+CCV*T3/3.—FCV/T
          H2=Psat*VG* 10.
          H3=A2/V1+A3/V2+A4/V3+A5/V4
          H4=C2/V1+C3/V2+C4/V3+C5/V4
          S1=ACV* log (T) + BCV* T + CCV* T2/2.—FCV/(2.* T2)
72        S2=10.* R* log (V1/(3.295/16.0185))
          S3=B2/V1+B3/V2+B4/V3+B5/V4
          S4=H4
73        H3=H3+A6* EMAV/alpha
          BM=EMAV/alpha
          S3=S3+BM* B6
          HG=H1+H2+10.* H3+10.* XKT1* (1.+XKT)* H4+XH
          SG=S1+S2—10.* S3+XS+10.*XKT1* XC*S 4/TCRIT
          HF=HG—HFG
74        SF=SG—SFG
75        Punch 105, T, PN, VF, VG, HF, HG, SF, SG
105       Format (2F 10.2, 2F 10.5, 2F 10.2, 2F 10.5)
          Go to 5
          Stop
          End

  0.96146700E—02     0.1248563600E—03    —0.1169819000E—01    0.1164312400E—04
 —0.1184098000      —0.2929524800E—05     0.2303194000E—07    0.2488960000E—03
  0.2419180000E—07  —0.6796653000E—10     0.0000000000       —0.2434407600E—10
  0.6301775700E—13  —0.1206109800E—09     0.4200000000E+01    0.3691800000E+03
  0.9400226200E+08  —0.2075806500E+09     0.8781341700E+04
524.76606      875.1612853      588.662575      —357.0934636      327.9513737
 27.18937909          —2136.218417          —7.86103122          0.0039435102
  0.445746703,           381.167               0.1137857608
0.1177678176,        0.001699729598,    —0.0000008830432915,      332.5417586
233.16
333.16
```

```
CC      Superheated vapour properties F - 22
C       Equation of state constants
        Read 101, R, B, A2, B2, C2, A3, B3, C3, A4, B4, C4, A5, B5, C5, XK, TCRIT, A6, B6,
        1alpba
101     Format (4E20.10/4E20.10/4E20.10/4E20.10/3E20.10)
100     Format (//)
        Punch 100
C       Specific heat constants
        Read, ACV, BCV, CCV, FCV
10      C1=2.3025851
        XH=34.66772* 4.1868
        XS=—0.00462254
12      Read, P, T
C       Calculate V
        TK=T
        PN=P
        E=2.71828** (—XK* T/TCRIT)
        E1=PN
        E2=R* T
        E3=A2+B2* T+C2* E
        E4=A3+B3* T+C3* E
        E5=A4+B4* T+C4* E
        E6=A5+B5* T+C5* E
        E8=0.0001
        E32=2.* E3
        E43=3.* E4
        E54=4.* E5
        E65=5* E6
        E7=A6+B6* T
C       Compute initial estimate of V from ideal gas law
        VN=R* T/PN
        Iter=0
C       Compute V by Newton iteration
1       Iter=Iter+1
        IF (iter—30) 2, 2, 20
2       V=VN
        V2=V* V
        V3=V2* V
        V4=V3* V
        V5=V4* V
        V6=V5* V
        EMAV=—alpha* (V+B)/2070.
        EMAV=2.71828**EMAV
        EMAV=EMAV**46.
        EMAV=EMAV** 45.0
        F=E1—E2/V—E3/V2—E4/V3—E5/V4—E6/V5—E7* EMAV
        FV=E2/V2+E32/V3+E43/V4+E54/V5+E65/V6+E7* alpha* EMAV
        VN=V—F/FV
        IF (ABS((VN—V)/V)—E8) 20, 20, 1
20      SVOLF=VN+B
        V=SVOLF
C       Calculate HG and SG
        T2=T* T
```

```
        T3=T2* T
        T4=T3* T
        V1=V—B
        V2=2.* V1* V1
        V3=3.* V1** 3
        V4=4.* V1** 4
        XKT=XK* T/TCRIT
        XKT1=2.71**(—XKT)
        EMAV=—alpha* V/2070.
        EMAV=2.71828**EMAV
        EMAV=EMAV**46
        EMAV=EMAV**45
55      H1=ACV* T+BCV* T2/2.+CCV* T3/3.—FCV/T
        H2=P* V* 10.
        H3=A2/V1+A3/V2+A4/V3+A5/V4
        H4=C2/V1+C3/V2+C4/V3+C5/V4
        S1=ACV* log(T)+BCV* T+CCV* T2/2.—FCV(2.* T2)
        S2=10.* R* log(V1/(3.2957/16.0185))
62      S3=B2/V1+B3/V2+B4/V3+B5/V4
        S4=H4
        H3=H3+A6* EMAV/alpha
        S3=S3+B6* EMAV/alpha
        H=H1+H2+10.*H3+10.*XKT1*(1.+XKT)*H4+XH
        S=S1+S2—10.* S3+XS+10.* (XKT1)* (XK)* S4/TCRIT
        Punch 99, P, T, V, H, S
99      Format (2F10.2, F10.5, F10 2, F10.5)
        Go to 12
        Stop
        End
    0.96146700E—02      0.1248563600E—03     —0.1169819000E—01    0.1164312400E—04
   —0.1184098000       —0.2929524800E—05      0 2303194000E—07    0.2488960000E—03
    0.2419180000E—07   —0.6796653000E—10      0.0000000000       —0.2434407600E—10
    0.6301775700E—13   —0 1206109800E—09      0.4200000000E+01    0.3691800000E+03
    0.9400226200E+08   —0.2075806500E+06      0.8781341700E+04
0.1177678176,      0.001699729598,      —0.0000008830432915,      332.5417586
0.69,    233.16
110.31616,     394.27
C  C    Superheated vapour properties F - 22
        .69       233.16      3.24130      235.88      1.27141
     110.32       394.27       .03218      334.40      1.11259
0       Error LC—2 in statement 0012+00 L. L.
```

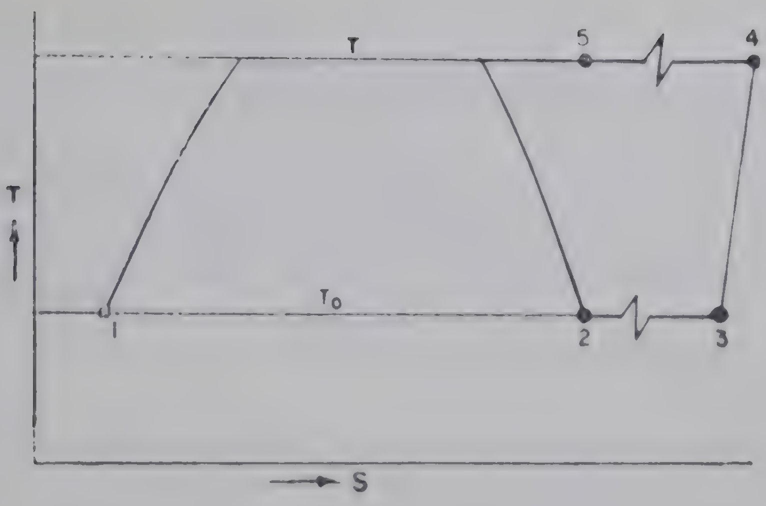

FIG. 1 T-S DIAGRAM

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Plotting of Exergy Diagram for Vapour Refrigerants

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The world has suddenly become aware of a power crisis of very severe nature. Steps are being taken to conserve energy resources and to ensure efficient use of electricity. Refrigeration equipment uses electricity. For a given refrigeration duty certain minimum electrical energy (work) W_m is required. However, the irreversibility of processes in various components of the equipment results in loss of work and hence the work actually required W_a is considerably more than the theoretical minimum. The exergy method gives the losses componentwise. Hence the method can be used to find the influence of various parameters on the losses in the components. Thus the working conditions can be optimized. However, exergy values for different states of the working substance must be readily available for this purpose. The values can be quickly obtained from an exergy chart. The paper discusses the method of plotting the exergy chart, the exergy enthalpy chart for vapour refrigerants. Data from refrigerant tables and p-h chart are used for plotting.

The exergy in a flow process is given by the expression.

$$e = h - T_o S + C$$

Where e, h and S are the exergy enthalpy and entropy per unit mass of the substance, while C is a constant and T_o the environment temperature.

T_o is assumed to be 300°K. The exergy of saturated liquid at 300°K is taken as zero and C is determined by using the values of h and s for the saturated liquid. Then the exergy values for saturated liquid and saturated vapour are calculated for temperatures from minus seventy to the critical point at intervals of 10°C. The saturation curve is plotted on the e–h chart. It is shown that the constant temperature and constant pressure lines are straight lines in the wet vapour region. Thus these lines are plotted.

The constant entropy lines are straight lines with slope equal to 1, the line $S = 1$ passes through the point representing saturated liquid at 0°C. An increment of $\triangle S = 0.1$ corresponds to an intercept of $\triangle h = 30$ kcal on a horizontal line. Thus constant entropy lines are plotted on the chart. The entropy and enthalpy values enable one to plot the constant temperature and constant pressure lines in the superheat region. The constant temperature lines drop downwards while the constant pressure lines are rising curves. The plotting of these lines completes the chart.

The chart can be used for finding the exergy values if $T_o = 300$°K. For other values of T_o a correction has to be added which is obtained from correction curves drawn at the bottom of the chart. The correction curve is a straight line for T_o = constant, on a chart for $\triangle e \; \alpha \; S$.

The exergy values are used in exergy balance equations to find the component-wise losses and the exergetic efficiency of the cycle under consideration.

Procedure

The chart is plotted for exergy in flow process of a vapour refrigerant. Exergy and enthalpy are shown on the Y axis and X axis respectively. Lines for constant entropy, constant pressure and constant temperature are also shown. Property tables and p-h chart are used for plotting.

Exergy in flow process is given by the expression

$$e = h - T_o S + C \qquad \ldots (1)$$

Take $T_o = 300$°K and p_o = the saturation pressure at 300°K. At p_o, T_o $e = e_o = 0$. Therefore

$$C = T_o S_o - h_o$$

where S_o, h_o are the property values for saturated liquid or saturated vapour at 300°K.

Thus for ammonia

$$C = 300 \times 1.1041 - 130.39 = 200.8$$
$$= 300 \times 2.0281 - 407.03 = 201.4$$
$$\therefore C = 201 \text{ correct upto 3 places}$$

and for ammonia

$$e = h - 300\,S + 201 \qquad \ldots (2)$$

Exergy of saturated liquid and vapour

Using equation 1 find out the exergy of saturated liquid and saturated vapour for various saturation temperatures from -70°C to the critical point at intervals of 10°C.

Similarly find the above two values for various saturation pressures at regular intervals (round figures for pressure).

For all cases note down the corresponding enthalpy values. Plot the saturation curve on the *e—h* chart, selecting suitable scale.

Exergy in wet region

$$e = h - T_{o} S + C$$

Therefore $de = dh - T_{o}\, ds$

$$\left.\frac{\delta e}{\delta h}\right)_p = 1 - T_{o} \left.\frac{\delta s}{\delta h}\right)_p$$

$$\text{But} \quad \left.\frac{\delta s}{\delta h}\right)_p = \frac{1}{T}$$

$$\therefore \quad \left.\frac{\delta e}{\delta h}\right)_p = 1 - \frac{T_{o}}{T} = \frac{T - T_{o}}{T}$$

in the wet region the temperature for a pure substance is constant if the pressure is constant. Therefore for the wet region

$$\left.\frac{\delta e}{\delta h}\right)_p = \frac{T - T_{o}}{T} = \text{constant}$$

or the constant pressure line, and hence the constant temperature line is a straight line in the wet region the slope being $T - T_{o}/T$.

Therefore join by straight lines the points representing the saturated liquid state and the saturated vapour state for a given temperature or pressure.

The constant temperature lines for temperatures above T_{o} have positive slope i.e. they slope upwards from left to right. Below T_{o} they slope downwards.

As seen above the slope of the constant pressure line is given by $T - T_{o}/T$. In the super heat region T increases and the slope increases with increase in T.

Constant Entropy Lines

$$de = dh - T_{o}\, ds$$

$$\text{Therefore} \left(\frac{\sigma e}{\sigma h}\right)_s = 1$$

Thus the constant entropy lines are straight lines with slope equal to 1.

Entropy of saturated liquid at O°C is equal to 1 $\frac{kcal}{kg°K}$

From this point draw a vertical downwards with $\Delta e = 10$ or any other convenient value. Then take a distance to the left $\Delta h = 10$ and thus draw the straight line $S = 1$.

$$e = h - T_o S + C$$
$$de = dh - T_o\, ds$$

for e = constant

$$\frac{dh}{ds} = T_o = \frac{\Delta h}{\Delta s}$$

or $\Delta s = \Delta h/300$.

On any horizontal line, $\Delta s = 0.1$ corresponds to $\Delta h = 30$. Thus draw the constant entropy lines throughout the range at an interval of $\Delta S = 0.1$ or any other suitable interval.

Thus in the superheat range we know the entropy and enthalpy values for all the points. Hence it is possible to plot the constant pressure and constant temperature lines in the superheat region with the help of data obtained from $p-h$ chart.

Nature of Various Curves/Constant Entropy Lines :

The constant entropy lines are parallel straight lines with slope = 1.

Constant Pressure Lines :

The constant pressure lines are straight lines in the wet region. The line po = constant is a horizontal line. All lines for $p < p_o$ are below the line po = const, and slope downwards in the wet region. The slope decreases with decrease in pressure. All the lines for p>po lie above the line po = const and slope upwards, the slope increasing with increase in pressure.

In the superheat region the line p = const for $p > p_o$ is a curve, for which the slope increases with increase in superheat, while for $p < p_o$ the curve passes through a minimum.

$$\left.\frac{\delta e}{\delta h}\right)_p = \frac{T - T_o}{T}$$

Hence for $T = T_o$ the slope is 0. Thus all the points corresponding to the minimum, lie on the line T_o = constant.

Constant Temperature Lines

$$\left.\frac{\delta e}{\delta h}\right)_T = 1 - T_o \left.\frac{\sigma s}{\sigma h}\right)_T$$

The constant temperature lines are straight lines in the wet region. Lines for $T < T_o$ slope downwards the slope decreasing with decrease in T, while for $T > T_o$ the lines slope upwards the slope increasing with increase in T.

In the superheat region if the pressure is low then the refrigerant may be assumed to behave like an ideal gas for which h = f (T). Hence at low pressures $\sigma h)_T$ = O and hence the slope is given by

$$\left.\frac{\delta e}{\delta h}\right)_T = 1 - \frac{T_o\, \sigma s}{O} = -\infty \quad \text{(infinity)}$$

Thus the constant temperature lines are almost vertical, slightly sloping towards the left in the low pressure region.

In the high pressure region the constant temperature line passes through a maximum given by the relation

$$\left.\frac{\delta e}{\delta h}\right)_T = O = 1 \quad T_o \left.\frac{\sigma s}{\sigma h}\right)_T$$

or at the maximum $\left.\frac{\sigma h}{\sigma s}\right)_T = T_o$

The above method may be used to plot the exergy diagram for NH_3, R12, R22 and other refrigerants.

Fig. 1 shows a typical diagram on which some sample curves are shown.

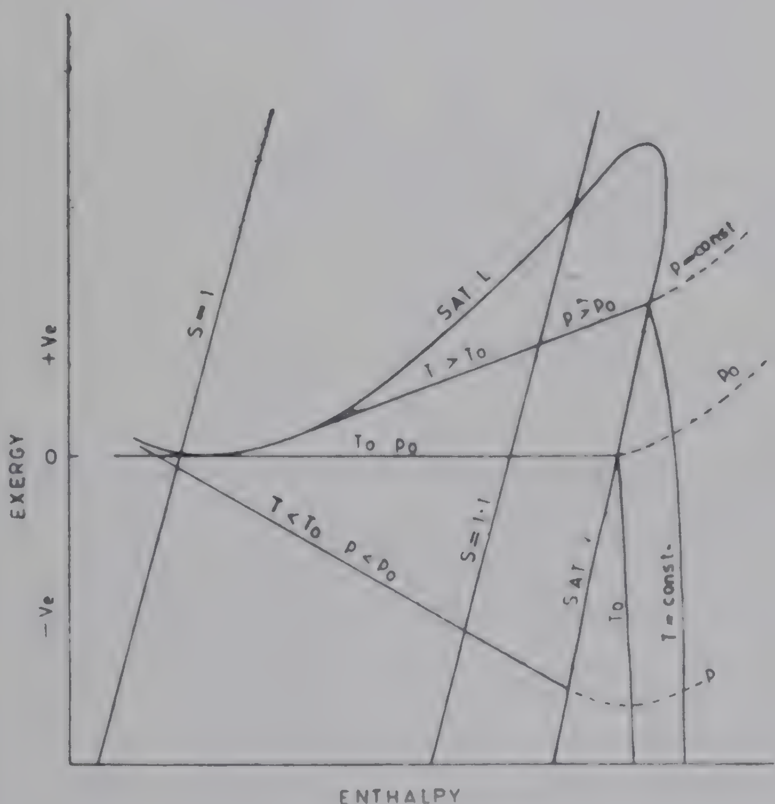

FIG ~~1/L 12~~ 1 - EXERGY ENTHALPY DIAGRAM FOR A VAPOUR

The diagram plotted as above will be useful for studying refrigeration cycle when the environment temperature is 300°K. The same diagram can be used for cases where the environment temperature is other than 300°K. For this purpose the exergy values obtained from the diagram will have to be modified.

*Exergy Values for $T_o \neq 300°$*K

The chart gives the absolute values of exergy for $T_o = 300°$K or $e_{300} = h - 300\ S + C_{300}$, for environment temperature T_o, we have

$$e_{T_o}' = h - T_o'\ S + C_{T_o}'$$

or $\triangle e = e_{T_o}' - e_{300} = (300 - T_o')S + C_{T_o}' - C_{300}$

$$= (300 - T_o')\ S + C$$

and $e_{T_o} = e_{300} + (300 - T_o')\ S + C$

The constant C gets eliminated when we consider exergy change between two states. In actual calculations we are interested in exergy changes only. Hence we can modify the exergy values obtained from the chart by adding to them the quantity $(300 - T_o')S$ and these modified values can be used for exergy analysis.

The following example illustrates the above.

Example : At points 1-2-3-4 the exergy and entropy values obtained from the chart are given in Table 1.

TABLE 1. *Modified Exergy Values*

Point	Exergy kcal/kg $T_o = 300°$K	Entropy kcal/kg °K	(300–320)S	(300–280)S	e_{320}–C	e_{280}–C′
1	+80	2.1	−42	+42	38	122
2	+10	0.9	−18	+18	− 8	28
3	—30	1.2	−24	+24	−54	−6
4	—140	2.5	−50	+50	−190	−90

Find modified exergy values if a) $T_o = 320°$K, b) $T_o = 280°$K.

a) $(300 - T_o')S = -20\ S = -42,\ -18,\ -24$ and -50 for the four points respectively. Hence we find e_{320}–C as $+ 80 - 42 = 38$ and so forth.

b) $(300 - T_o')S = 20\ S$, hence we calculate the other values. The calculated values are shown in the table,

Instead of calculations it is possible to draw curves at the bottom of the charts showing $\triangle e \propto S$.

Conclusion

The method of exergy balance gives the losses in a system componentwise. For finding out the losses we need the exergy values at various state points. The exergy chart gives these values. Hence the exergy chart is extremely useful for thermodynamic analysis. The paper discusses the method of plotting an exergy chart for a vapour refrigerant,

Regeneration And Refrigeration

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Introduction

For a given range of temperature there is no engine cycle which is more efficient than the carnot cycle. Similarly in refrigeration for maintaining a space at a given temperature no cycle is more efficient than the reversed carnot cycle. A layman would therefore expect all engines to work on carnot cycle and most of the refrigerating machines to work on the reversed carnot cycle. However an engineer knows that not a single machine works on these cycles. What is the reason? What is it that makes the carnot cycle the most efficient? What is it that makes it impracticable?

The working substance for engines and refrigerators is a gas or a vapour. The working substance passes through a range of temperature. Thus for the I. C. engines the maximum temperature is around 2500°K, for gas turbines it is around 1200°K and for steam plants it is about 900°*K*. The minimum temperature corresponds to the environment temperature. For refrigeration the environment receives the rejected heat and the refrigeration temperature may have any lower value depending upon the requirements.

The carnot engine receives heat at the highest temperature and rejects it at the lowest temperature. The carnot refrigerator also receives heat at as high a temperature as permissible and rejects heat at as low a temperature as permissible. This results in the high efficiency of the carnot cycle. Thus the secret of the high efficiency of the carnot cycle lies in the heat exchange at the extreme ends of the working range. The cycle is completed by having adiabatic compression and expansion between the two extreme temperatures. During a reversible adiabatic process there is a rigid connection between the temperature ratio and the pressure ratio.

For a gas the relation is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\nu-1}{\gamma}} \qquad 1$$

where γ is the adiabatic index. For the wet region of a vapour $\ln p = 1/T+C$ gives a simple relation between p and T. Thus for carnot cycle for air with a moderate isothermal compression ratio of say 5, the total compression ratio is of the order of a few hundred for a temperature ratio of just 3 (i.e. 300° to 900°K or 100°K to 300°K). For the temperature ratios that we come across in the I.C. engines the pressure ratio will be a few thousand.

For steam in the wet region the pressure ratio is about 4000 for a temp. ratio of 2. For R12—R13 and R22 it is about 1000 for a temperature ratio around 2. For ammonia also it is of the same order (Fig. 1.)

Thus we see that the carnot cycle becomes impracticable because of the high pressure ratios encountered.

One of the solutions is to have adiabatic compression to an intermediate temperature and then to have heat transfer through a range of temperature, as is done in the Joule, Diesel and Otto cycles. However this results in lowering the efficiency. The efficiency is further reduced and that too considerably if the adiabatic processes are irreversible.

For large capacities it becomes necessary to use turbomachines which are very compact on account of their speeds which are several times those of the reciprocating machines. However for a turbomachine the pressure differential has to be very low if the lower pressure is limited to 1. This results in a very low efficiency of the simple turbomachine as the temperature range through which the heat is supplied increases appreciably.

Refrigeration Cycles

If air is used as the working substance to maintain 100°K with the environment at 300°K the carnot cycle will require a pressure ratio of about 250. If the carnot

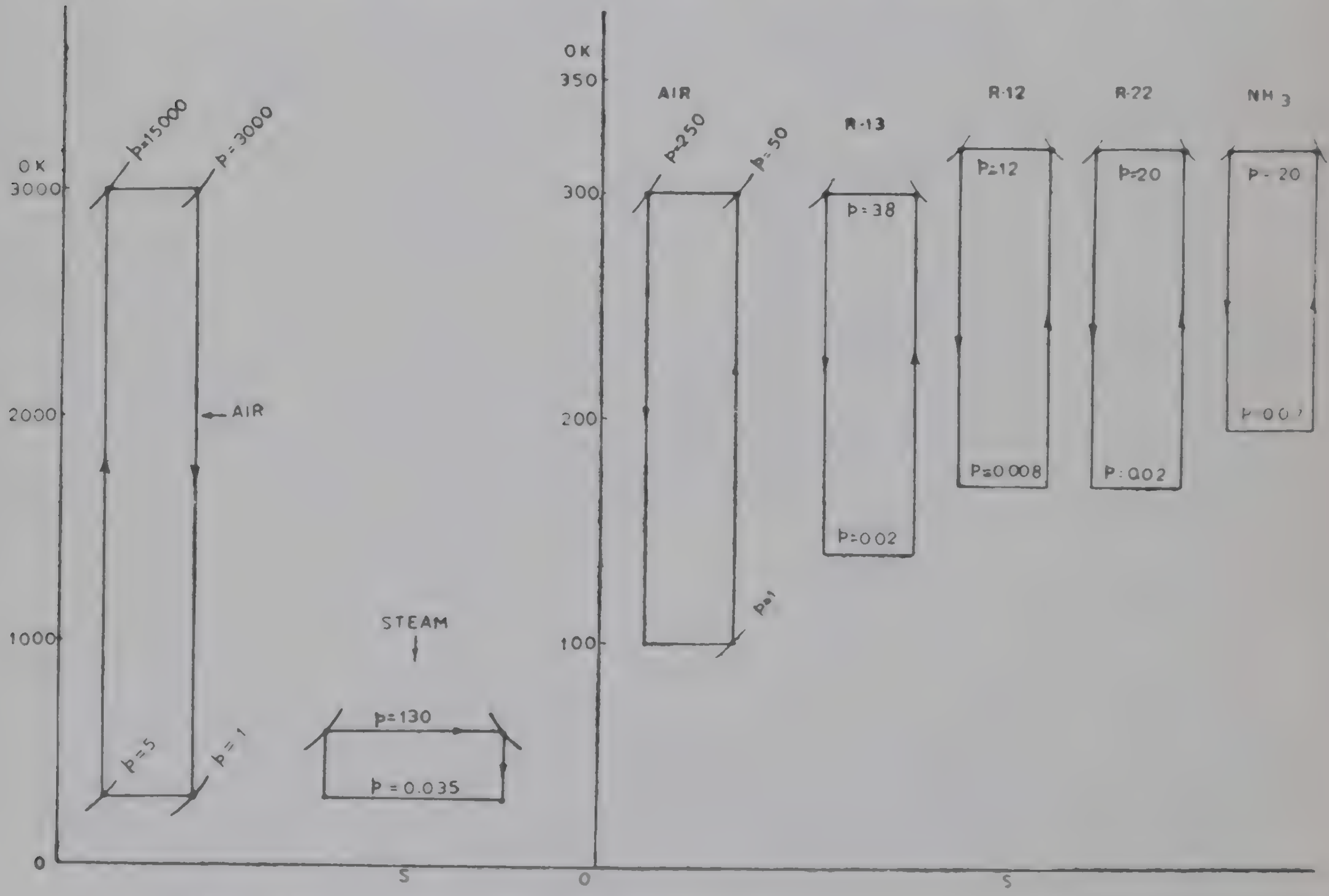

FIG.1—CARNOT CYCLE FOR ENGINES AND REFRIGERATORS

cycle is replaced by the Bell Coleman cycle with $\Delta T = 40°$ the pressure ratio required will be the same, however the efficiency will by only 50 per cent of the carnot machine. Further the actual machine will not attain the temperature of 100°K unless the expander efficiency is more than 83 per cent. Usually the expander efficiency is around 70 per cent, which gives lowest temperature of 132°K only.

For vapour compression machines a pressure ratio of about 800 will be required to produce temperatures around 175°K if the environment is around 310°K. The temperature at the end of compression will also be very high. The compressor volumetric efficiency reduces to zero for a clearance volume of 3 per cent if the pressure ratio is more than 100,60 or 50 respectively for NH_3, R—22 and R12. Hence for a ratio higher than the above mentioned the compressor will not suck in any vapours. The throttle valve losses increase, thus resulting in the reduction of specific refrigerating effect and reduction in efficiency. The excessively large specific volume at low pressures results in considerable reduction in the refrigerating capacity of a given machine.

Thus we find that both the gas cycle and the vapour cycle become impracticable for producing low temperatures as the pressure ratios involved are very high. We therefore have to consider a method which gets rid of the rigid connection between the temperature ratio and the pressure ratio. We should be able to change the temperatures without appreciably changing the pressures or pressure ratios. Regeneration is the solution.

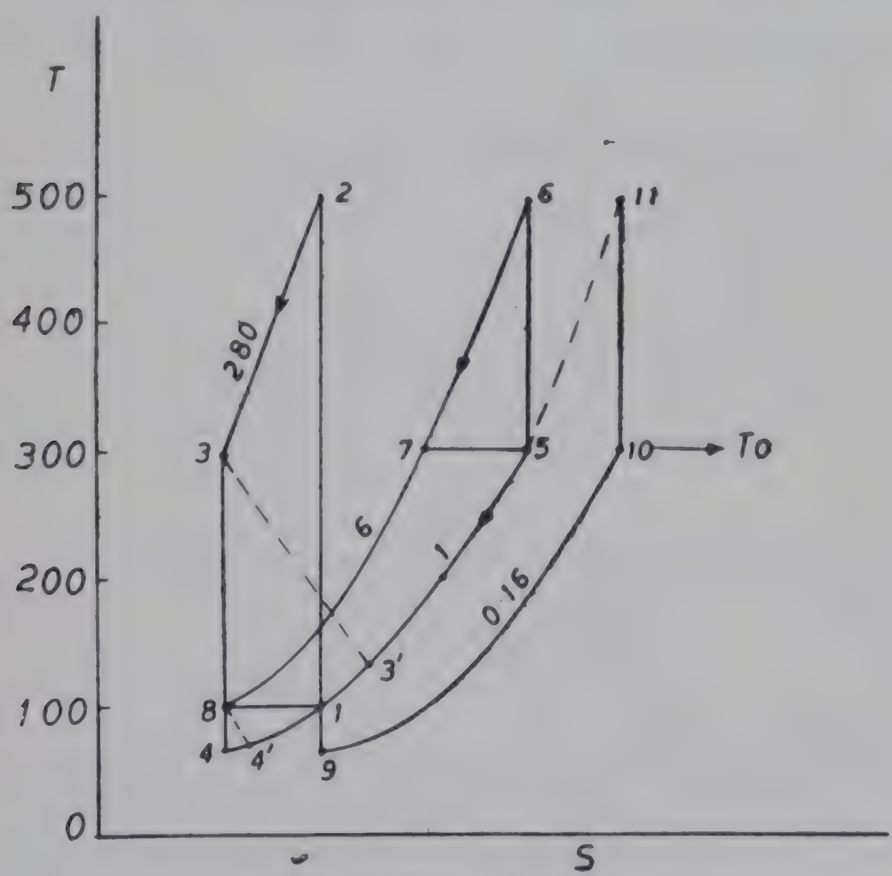

FIG. 2. Air Refrigeration Cycles

Regeneration

Regeneration is internal heat exchange, heat exchange between two components of the system or between two internal streams. If a stream exchanges heat at constant pressure then its temperature can be varied without any change in pressure In a constant volume process the temperature can be varied with only a moderate change in pressure.

Regeneration and the gas cycle

In Fig. 2. 1—2—3—4—1 represents the above mentioned Bell Coleman cycle. 5—6—7—8—4—1—5 represents a regenerative cycle having the same efficiency and a max. pressure of 6 only in place of 280. 5—7—8—1—5 represents a regenerative cycle the reversed Ericsson cycle which is as efficient as the carnot cycle. 5—4—1—9—10—11 represents the Martinovsky cycle which works with maximum pressure of 1 atm only and a pressure differential of 0.84. Cycle 5—6—7—8—4′—1—5 is a regenerative cycle with the expander having an efficiency of say 70 per cent. Here we see that unlike the Bell Coleman cycle, that cycle is capable of producing the desired temperature. Only the efficiency of this cycle is somewhat lower than that of the theoretical regenerative cycle.

Regeneration and the Linde Machine

The Linde machine uses a throttle valve as an expansion device. In throttling air from 200 atm to 1 atm a temperature drop of about 35°C only is produced. However a regenerator enables us to utillze the refrigeration at a much lower temperature thus producing liquefaction of air.

The Vapour Compression Machine and Regenerntion

Fig. 3 shows the use of regeneration for vapour compression systems. The figure represents a multistage system or a cascade system. The multistage system

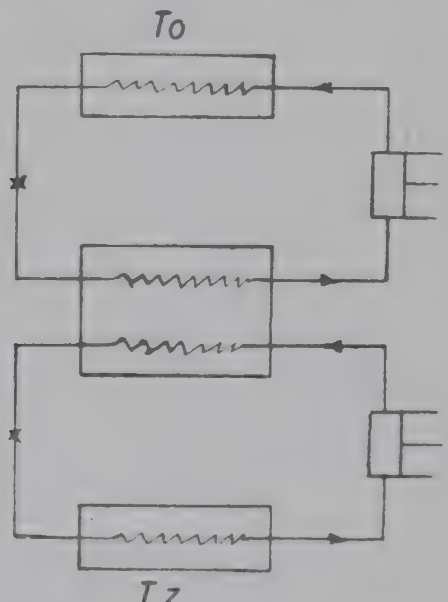

FIG. 3. Cascade System

results in smaller pressure ratio per stage thus improving the volumetric efficiency. The maximum temperature is reduced and there is an improvement in efficiency as well. In cascade system a low boiling substance can be used in the lower cascade, thus making it possible to increase the refrigerating capacity per unit volume.

Conclusion

It is impossible to attain low temperatures in refrigeration without the use of regeneration. Regeneration makes it possible to use turbomachines. It enables us to liquefy air. With the help of regeneration we get a practicable refrigeration cycle which is as efficient as the carnot cycle.

Irreversibility And Refrigerating Machines

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Introduction

According to the second law of thermodynamics input of work is a must for producing refrigeration. Expenditure of compression work is a common method in refrigeration. The working substance is a gas or a vapour. The gas may some times behave like an ideal gas. To obtain continuous refrigeration the working substance is made to perform a cyclic process i.e. the process is repeated over and over again. A refrigeration system consists of a number of parts, a number of components. Every compression system needs certain basic components. In addition some accessories are used.

For a particular refrigeration duty certain minimum work is required. The minimum work corresponds to the work required by the most ideal process. In an actual plant all the processes are irreversible. In every irreversible process there is a loss of work. Hence in each component there will be some loss of work. And hence the actual work required by a refrigeration system for a given duty will be equal to the minimum work plus the sum of the losses of work in all the components or we may write :

$$W_a = W_{min} + \Sigma W_l \qquad \ldots\ldots \quad (1)$$

The ratio of the minimum work to the actual work is known as the exergetic efficiency or

$$\eta_e = \frac{W_{min}}{W_a} = \frac{W_{min}}{W_{min} + \Sigma W_l} = \frac{W_a - \Sigma W_l}{W_a} \qquad \ldots\ldots \quad (2)$$

We discuss below some of the methods for finding out the losses, component-wise and hence obtaining the efficiency.

We often use the coefficient of performance to represent the performance of a refrigeration system the COP is given by the equation

$$COP = \frac{\text{refrigerating effects}}{\text{work input}} = \frac{q_z}{w_a} \qquad \ldots\ldots \qquad (3)$$

Hence we may write an equation for efficiency in terms of the *COP*

$$\eta_e = \frac{W_{\min}}{W_a} = \frac{q_z}{COP \text{ ideal}} \div \frac{q_z}{COP \text{ actual}}$$

$$\eta_e = \frac{COP_a}{COP_i} \qquad \ldots\ldots \qquad (4)$$

After finding the losses component-wise we express them as a fraction of the actual work input. Thus we find whether the losses are appreciable or otherwise in a particular component.

Minimum Work

In refrigeration we come across two types of requirements (a) Removing a quantity of heat q_z from a body or space in order to maintain it at a temperature T_r. (b) Removing a quantity of heat q_z from a body or a stream in order to obtain it in a suitable state at a temperature T_z, lower than the environment temperature T_o, e.g. producing ice or liquid air.

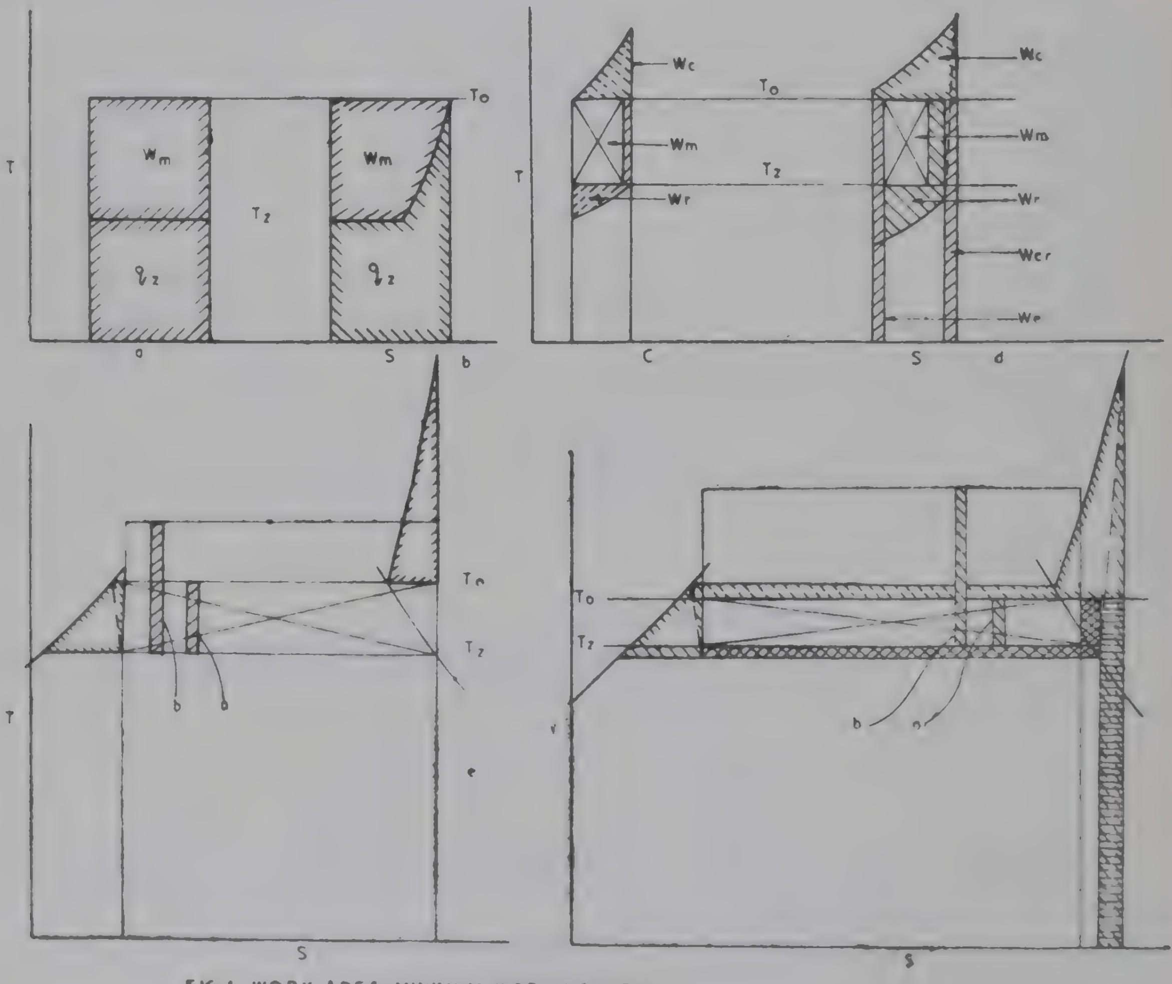


FIG. 1 - WORK AREA - MINIMUM WORK - LOSSES

Fig. 1a show the minimum work for type a and 1b shows the minimum work for type b.

Work Area Method

The actual work required by a refrigeration system can be shown on the T-s diagram under certain assumptions. The area can be further divided into W_{min} and W_l for each component.

Fig. 1e shows a Bell Coleman cycle consisting of two reversible adiabatic processes and two constant pressure processes which are externally irreversible. The actual work W_a will consists of three component W_{min}, W_{lr} the work lost in the refrigerator and W_{le} the work lost in the cooler.

Fig. 1d shows a Bell Coleman cycle for which all the four processes are irreversible. The actual work area is divided into 5 parts. The areas represent the work lost in the expander, W_e; refrigerator, W_r; compressor, W_{lr}; and the cooler, W_c respectively and minimum work, W_m.

Similarly Fig. 1e represents the simple vapour compression cycle. The processes in the evaporator and the compressor are assumed to be reversible. In the condenser the desuperheating takes place at finite temperature difference hence the process is assumed to be externally irreversible while the throttling process is internally irreversible. The actual work area is divided into three parts, the minimum work, the work lost in the condenser, the work lost in the throttle valve.

For the cycle of Fig. If all the four processes are irreversible the area of figure represents the actual work. The five parts of the area represent the minimum work and the work lost in each of the four components.

If the above diagrams are drawn to scale we get an idea about the relative importance of the loss in each components.

Equivalent Carnot Cycle COP and the Relative Efficiency

For the carnot cycle the COP is expressed in a simple way in terms of temperatures. Every refrigerating machine receives and rejects heat and the work input is equal to the difference between the heat rejected and the heat received. If the two heat quantities are shown as rectangular areas with the same width then we get a carnot cycle which has the same COP as the COP of the given cycle. This also enables us to show the exergetic efficiency on the T–s diagram. It will be the ratio of the height of the two carnot cycles one representing the ideal cycle and the other representing the actual cycle, both producing the same amount of refrigerating effect.

Thus figure 2A shows an equivalent carnot cycle. B and C show the diagramatic representation of η_e for the air cycles discussed earlier. The efficiency being given by the expression

$$\eta_e = a/b \qquad \ldots\ldots\ldots \qquad (5)$$

The efficiency of the vapour cycles is shown on Fig. 1.

The Entropy Increase Method

According to the second law of thermodynamics the entropy of an isolated system increases during an irreversible process. There is a loss of work given by

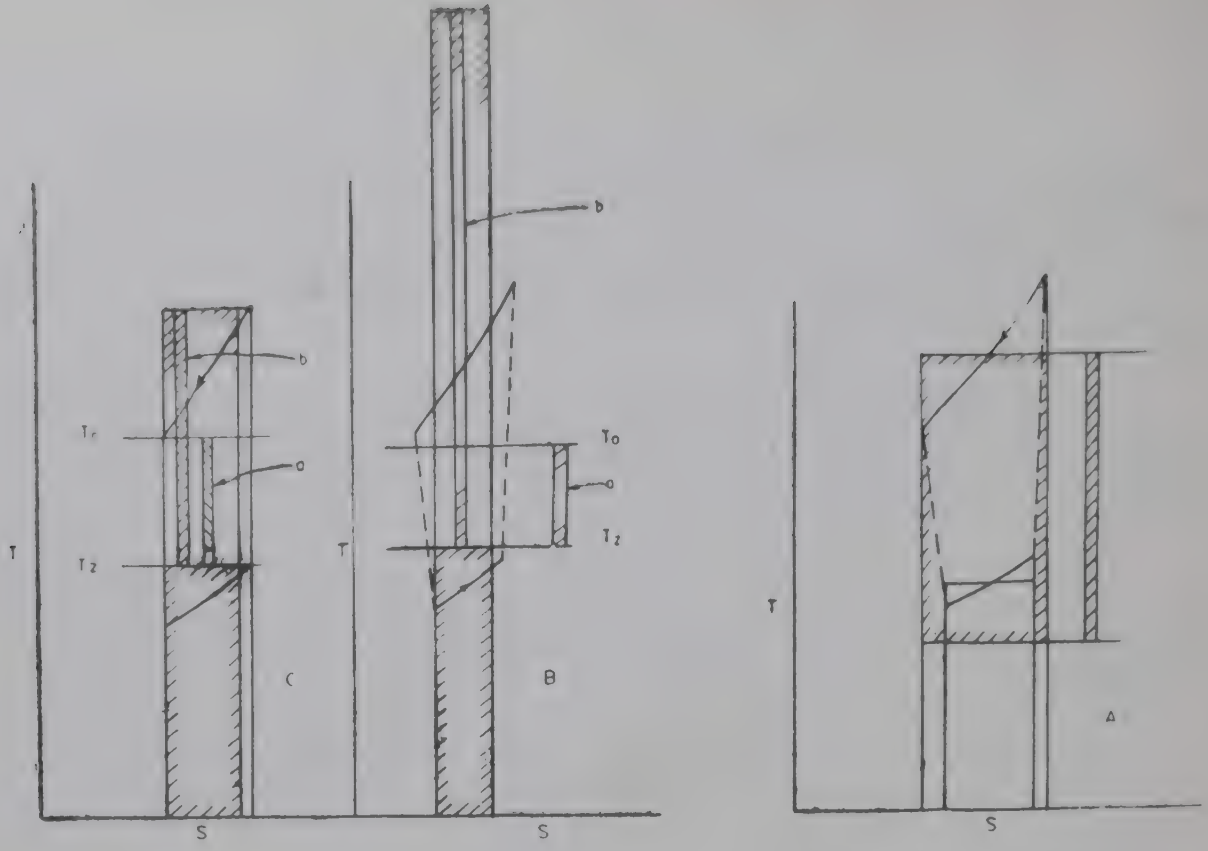
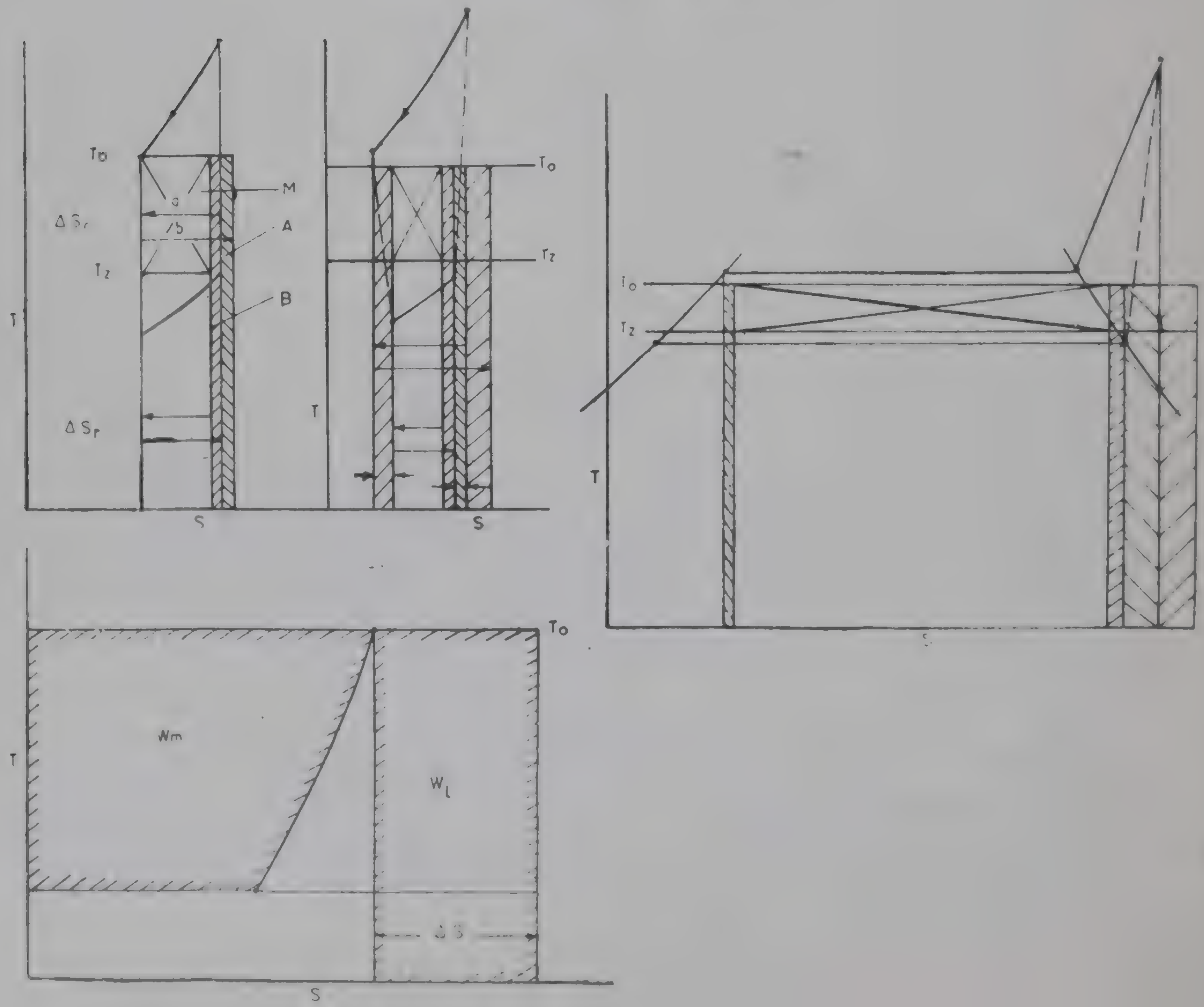


FIG-2. Eq. CARNOT CYCLE AND EFFICIENCY



$$W_1 = \triangle S \; T_o$$

Where $\triangle S$ is the increase of entropy in the irreversible process and T_o is the environment temperature. To find W_1 for any component we treat the component and its surroundings as an isolated system and find the increase in entropy of the isolated system during the process under consideration. Thus in Fig. 3, a represents the decrease in entropy of the air in the cooler and b represents the increase in entropy of the environment and the shaded area A gives the loss in the cooler. Similarly B gives the loss in the refrigerator and M the minimum work. The losses in the other cases also are represented in the figure.

Carnot cycle is not the ideal cycle for liquefaction of air. Fig. 3d shows the minimum work and the loss of work when carnot cycle is used for air liquefaction.

The Exergy Method

The exergy of a system in flow process is given by the equation

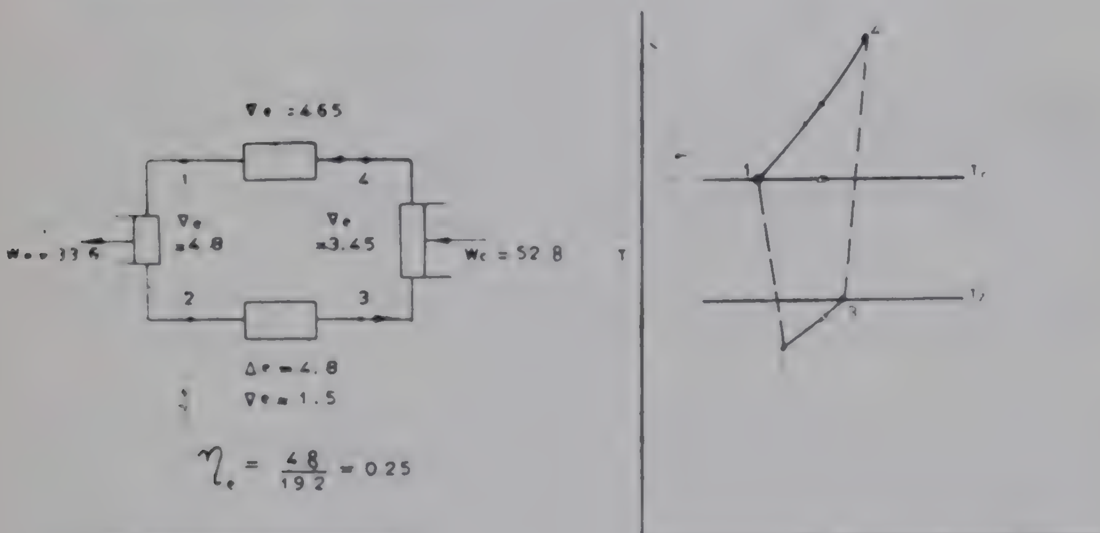
$$e = h - T_o \, S - (h_o - T_o \, S_o) \qquad \ldots\ldots\ldots\ldots \qquad (6)$$

In every real process there is a loss of exergy in a component the loss of exergy ∇e is found by exergy balance for the component. Thus

$$\nabla e = e_1 + e_r - e_g - e_2 \qquad \ldots\ldots\ldots\ldots \qquad (7)$$

Where e_1 = exergy of incoming stream

e_2 = exergy of leaving stream



STATE	T	h	e
1	300	33.6	54.6
2	160	0.0	16.2
3	200	9.6	9.9
4	20	62.4	59.25

FIG. 4. **Exergy Balance**

e_r = exergy received by the stream in the component due to addition of heat or work

e_g = exergy given away by the stream.

The exergy values may be calculated with the help of equation No. 6 or they may be found out from an exergy chart.

Fig. 4 shows a Bell Coleman cycle and the losses in the various components found with the help of exergy chart.

Conclusion

For a refrigeration system the actual work is more than the minimum required due to the irreversibility in various components. It is necessary to know the losses component-wise, to note the relative importance of the losses and to find out the effect of the losses on the overall performance. The paper discusses four different methods of analysis of the losses.

Transient Process of Freeze-Drying of A Semi-Infinite Slab

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The paper describes the general freeze-drying problem considering the transient heat transfer process of a semi-infinite slab with combined heating by radiation and conduction from one side and by contactive conduction from the other side. A set of equations are established for heat transfer through the dried, frozen and external regions and are solved simultaneously to give an analytical solution for the temperature distribution.

Also, calculations are made for a typical case with nitrogen present as an inert gas from the numerical data available in literature.

Introduction

Freeze-drying is a successful process of liquid separation from a product, achieved by sublimation. Mostly, it is used for dehydration.

One direction, in which work needs to be done, is the improvement of the physical process.

Investigations of the phenomena, parameters and properties involved in it, for which there is still lack of data, are required in order to choose their reasonable values and inter-relations for optimal designing and control.

Theoretical Model

There is need to establish an optimal conceptual system for predicting the physical process, expressible by mathematical model. For this reason, the following conditions have been considered :—

a) One dimensional semi-infinite slab,

b) Unsteady state heat transfer,

c) Combined heating both through the frozen and dried region which is suggested to be accelerating for immovable beds[4],

d) Combined heat and mass transfer,

e) External heat and mass transfer[6–9],

f) Combined radiative and conductive external heating in the presence of gas. suggested to accelerate the process,[5]

g) The mode of mass flow both under the total pressure and concentration gradient driving forces, on the basis of molecular theory of fluids,

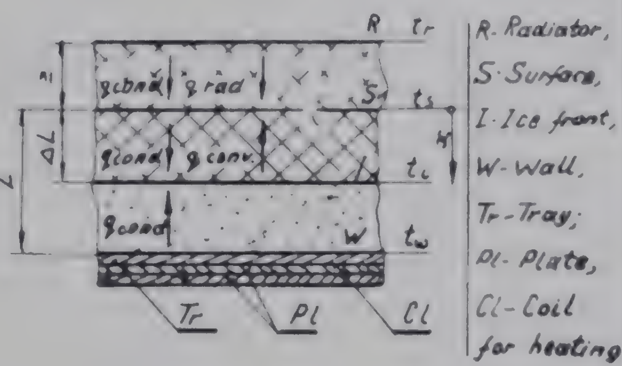


fig1.
Semi-infinite slab
under freeze-drying

h) Convection due to the water vapour passing through the dried layer, and

i) Generalization of the analytical solutions by introducing dimensionless forms of equations.

The process seems to be heat transfer governed at low pressures. The attention is given, therefore, to the heat transfer.

The equations describing the temperature fields are established and solved analytically for suitable boundary conditions. Numerical example for a set of data is given.

Set of equations

The temperature fields are described by the following equations (Fig 1):

(a) Dried region for conduction and convection[3]

$$\frac{\delta^2 t_d}{\delta x^2} + \frac{N_w C_p}{\lambda_d} \cdot \frac{\delta t_d}{\delta_x} = \frac{1}{\alpha_d} \cdot \frac{\delta t_d}{\delta \tau} \tag{1}$$

(b) Frozen region for conduction only

$$\frac{\delta^2 t_f}{\delta x^2} = \frac{1}{\alpha_f} \cdot \frac{\delta t}{\delta \tau} \tag{2}$$

(c) External space (chamber)

$$\frac{\delta^2 t_e}{\delta x^2} = \frac{1}{\alpha_e} \cdot \frac{\delta t_e}{\delta \tau} \tag{3}$$

The boundary energy balance equation for $\tau > o$ are

(a) At the surface S (Fig. 1), x=0

$$-\lambda d \frac{\delta t_a}{\delta x}\bigg|_{\substack{x=0 \\ \tau>0}} - \lambda e \frac{\delta t_e}{\delta x}\bigg|_{\substack{x=0 \\ \tau>0}} = h_r(t_r - t_s) \tag{4}$$

where from [5]

$$h_r = \frac{4 \ \sigma_r \ (T_{avg})^3}{\frac{1}{\epsilon_r} + \frac{1}{\epsilon_s} - 1} \tag{5}$$

and

$$T_{avg} = \frac{T_r + T_s}{2}$$

(b) At the ice front:

$$-\lambda_d \frac{\delta t_d}{\delta x}\bigg|_{\substack{x=\triangle L \\ \tau>0}} + \lambda f \frac{\delta t_f}{\delta x}\bigg|_{\substack{x=\triangle L \\ \tau>0}} = \triangle H_s, N_w \tag{6}$$

It can be understood that at $\tau = 0$, L=0 which are the initial boundary conditions.

Now N_w can be developed to give:

$N_w = \varrho_i \cdot \sigma . u$

$$\text{where } u = \frac{d(\triangle L)^{4,10}}{d\tau} \tag{7}$$

Now (7) becomes

$$N_w = \varrho_i \, . \, \sigma \, . \, \frac{d\,(\triangle L)}{d\tau} \tag{8}$$

Using linear relation between the ice front position and the change of moisture content[10]:

$$L=(1-X)\,.\,L \tag{9}$$

and substituting it in Equation (8) the form of equations (1) and (6) are changed as follows:

$$\frac{\delta^2 t_d}{\delta x^2} + \frac{\varrho_i \, . \, \sigma \, . \, C_p}{\lambda d} \, . \, \frac{d\,(\triangle L)}{d\tau} \, . \, \frac{\delta t_d}{\delta x} - \frac{1}{\alpha_d}\frac{\delta t_d}{\delta \tau} = 0 \qquad \cdots (1)$$

$$-\lambda d \frac{\delta t_d}{\delta x}\bigg|_{\substack{x = \triangle L \\ \tau > 0}} + \lambda f \frac{\delta t_f}{\delta x}\bigg|_{\substack{x = \triangle L \\ \tau > 0}} = \triangle H_s \, \varrho_i \, \sigma \frac{d(\triangle L)}{d\tau} \qquad \cdots (6)$$

Solution of the set of equations

Analytical solution of Eq. (1) is possible only by knowing the analytical expression for $\frac{d\,(\triangle L)}{d\tau}$. An 'apriori' approximation is suggested by Newmann for heat transfer as follows:

$$\triangle L = b \, . \, \sqrt{\tau} \tag{10}$$

Introducing it into equations (1) and (6) along with C_1 defined here as

$$C_1 = \frac{\varrho_i \, . \, \sigma \, . \, C_p}{2 \; \lambda_d} \tag{11}$$

the equations (1) and (6) will change to

$$\frac{\delta^2 \; t_d}{\delta x^2} + \frac{C_1 \, . \, b}{\sqrt{\tau}} \, . \, \frac{\delta t_d}{\delta x} - \frac{1}{\alpha d} \, . \, \frac{\delta t_d}{\delta \tau} = 0 \tag{1}$$

and

$$-\lambda_d \frac{\delta t_d}{\delta x}\bigg|_{\substack{x = \Delta L \\ \tau > 0}} + \lambda_f \frac{\delta t_f}{\delta x}\bigg|_{\substack{x = \Delta L \\ \tau > 0}} = \Delta H_s \, . \, \varrho_i \, . \, \sigma \frac{d}{2\sqrt{\tau}} \tag{6}$$

where the value of b remains undetermined.

Similarly transformation to dimensionless form is now used as is shown below:

a) For the temperatures

$$t_d^* = \frac{t_d - t_i}{t_s - t_i} \tag{12}$$

$$t_f^* = \frac{t_f - t_i}{t_w - t_s} \tag{13}$$

$$t_e^* = \frac{t_e - t_s}{t_r - t_s} \tag{14}$$

(b) For the time

$$\tau_d^* = \frac{\alpha_d \cdot \tau}{L^2}; \quad \tau_e^* = \frac{\alpha_e \cdot \tau}{L^2}; \quad \tau_f^* = \frac{\alpha_f \cdot \tau}{L^2} \tag{15}$$

c) For the distance

$$x^* = \frac{x}{L} \tag{16}$$

The dimensionless forms of the equations become :

$$\frac{\delta^2 \; t_d{}^*}{\delta x^{*2}} + \frac{C_2 \, , b}{\sqrt{\tau_d{}^*}} \cdot \frac{\delta t_d{}^*}{\delta x^*} - \frac{\delta t_d{}^*}{\delta \tau_d{}^*} = O \tag{1''}$$

$$\frac{\delta^2 \; t_f{}^*}{\delta x^{*2}} - \frac{\delta t_f{}^*}{\delta \tau_f{}^*} = O \tag{2'}$$

$$\frac{\delta^2 \; t_e{}^*}{\delta x^{*2}} - \frac{\delta t_e}{\delta \tau_e} = O \tag{3'}$$

$$- \frac{\delta t_d{}^*}{\delta x^*}\Bigg|_{\substack{x^* = O \\ \tau_d{}^* > O}} = \frac{t_r - t_s}{t_s - i_i} \cdot \frac{l}{\lambda_d} \left[L \, . \, h_r + \lambda_e \Bigg|_{\substack{x^* = O \\ \tau_e{}^* > O}} \right] \tag{4'}$$

$$- \lambda_d (t_s - t_i) \frac{\delta t_d{}^*}{\delta x^*}\Bigg|_{\substack{x^* = \frac{\Delta L}{L} \\ \tau_d{}^* > O}} + \lambda_f \, (t_w - t_i) \; \frac{\delta t_f{}^*}{\delta x^*}\Bigg|_{\substack{x^* = \frac{\Delta L}{L} \\ \tau_f{}^* > O}}$$

$$- \frac{\Delta H_s}{C_p} \cdot \frac{C_2 \, b \, \lambda_d}{\sqrt{\tau_d{}^*}} = O \tag{6''}$$

where

$$C_1 \sqrt{\alpha_d} = C_2 \tag{17}$$

Solutions of Equation (1")

Using the linearising substitution :

$$\eta_d = \frac{x^*}{2\sqrt{\tau_d{}^*}} \tag{18}$$

we obtain

$$\frac{\delta^2 t_d}{\delta \eta^{2d}} + 2 \, (C_2 \, b + \eta_d) \; \frac{\delta t_d{}^*}{\delta \eta_d} = O$$

Similarly transformation is possible with

$$C_2 \; b \; + \; \eta_d \; = \; u \tag{19}$$

which transforms the equations into an ordinary differential equation :

$$\frac{\delta^2 t_d}{\delta u^2} + 2u \; \frac{\delta t_d{}^*}{\delta u} = O$$

The analytical solution is :

$$t_d{}^* = C_o \int_O^u e^{-u^2} \; du \; + \; C_5 \tag{19'}$$

Letting C_o to be constituted by C_4 and $\frac{2}{\sqrt{\pi}}$

$$t_d^* = C_4 \frac{2}{\sqrt{\pi}} \int_o^u e^{-u^2} du + C_5 \qquad (19'')$$

in which $\frac{2}{\sqrt{\pi}} \int_o^u e^{-u^2} du$ is the error function.

Thus the solution becomes:

$$t_d^* = C_4 \, erf\, u + C_5.$$

In developed form this is:

$$t_d^* = C_4 \; erf \; (C_2 \; b + \eta_d) + C_5$$

or

$$t_d^* = C_4 \; erf \; (\frac{x^*}{2\sqrt{\tau_d^*}} + C_2 \; b) + C_5$$

Two additional boundary conditions are required for defining C_4 and C_5;

$$t_d^* = 1 \text{ at } x^* = O \text{ and } \tau_d^* > O \qquad (20)$$

$$t_d^* = O \text{ at } x^* = \frac{\Delta L}{L} \text{ and } \tau_d^* > O \qquad (21)$$

The final form of the solution for t_d^* is thus obtained

$$t_d^* = \frac{erf\,(\frac{\Delta L}{2L\sqrt{\tau_d^*}} + C_2\, b) - erf\,(\frac{x^*}{2\sqrt{\tau_d^*}} + C_2\, b)}{erf\,(\frac{\Delta L}{2L\sqrt{\tau_d^*}} + C_2\, b) - erf\,(C_2\, b)} \qquad (22)$$

The derivative of t_d^* for the dimensionless time is:

$$\frac{\delta t_d^*}{\delta x} = \frac{\frac{-2}{\sqrt{\pi}} \, exp\,[-(\frac{x^*}{2\sqrt{\tau_d^*}} + C_2\, b)^2]}{erf\,(\frac{\Delta L}{2L\sqrt{\tau_d^*}} + C_1\, b) - erf\,(C_2\, b)} \qquad (23)$$

and at the interface

$$\left.\frac{\delta t_d^*}{\delta x^*}\right|_{\substack{x^* = \frac{\Delta L}{L} \\ \tau^*_d > O}} = \frac{-\frac{2}{\sqrt{\pi}} \, exp\left\{-\left[b\left(\frac{1}{2\sqrt{\alpha_d}} + C_2\right)\right]^2\right\}}{erf\,[\,b\,(\frac{1}{2\sqrt{\alpha_d}} + C_2) - erf\,(C_2 b)]}$$

Solution of Equation (2')

Using similar transformation formula $\eta_t = \frac{x^*}{2\sqrt{\tau_t^*}}$ as in Equation (18) we obtain the equation

$$\frac{\delta^2 t_t^*}{\delta \eta_t^2} - 2n_t \frac{\delta t_t^*}{\delta \eta_t} = 0, \text{ for which the solution is } t_t^* = C' \operatorname{erf} \eta_t + C'' \qquad (26)$$

or

$$t_t^* = C' \operatorname{erf} \frac{x^*}{2\sqrt{\tau_t^*}} + C''$$

The solution at the interface is

$$\left.\frac{\delta t_f^*}{\delta x^*}\right|_{\substack{x^*=\frac{\triangle L}{L} \\ \tau_f^*>0}} = \frac{-\frac{2}{\sqrt{\pi}} \exp\left[-\left(\frac{b}{2\sqrt{\alpha_f}}\right)^2\right]}{erf\left(\frac{b, L}{2\triangle L\sqrt{\alpha_f}}\right)-\mathrm{erf}\left(\frac{b}{2\sqrt{\alpha_f}}\right)} \tag{27}$$

Solution of Equation (3′)

Similar substitution as in Equation (18) $\eta_e = \frac{x^*}{2\sqrt{\tau_e^*}}$ is valid and we obtain the solution.

$$t_e^* = C^+ \, erf\, \eta_e + C^{++}$$

$$t_e^* = C^+ \, erf\left(\frac{x^*}{2\sqrt{\tau_e^*}}\right) + C^{++}$$

Boundary equations are :

$$t_e^* = 1 \text{ at } x^* = -\frac{Z}{L} \text{ and } \tau_e^* > 0 \tag{28}$$

and

$$t_e^* = 0 \quad \text{at} \quad x^* = 0 \quad \text{and} \quad \tau_e^* > 0 \tag{29}$$

The solution for the external temperature distribution equation is thus

$$t^*_e = \frac{erf\left(\frac{x^*}{2\sqrt{\tau_e^*}}\right)}{erf\left(-\frac{Z}{2L\sqrt{\tau^*_e}}\right)} \tag{30}$$

The derivative with respect to the distance is :

$$\frac{\delta t_e^*}{\delta x^*} + \frac{2/\sqrt{\pi} \quad \exp\left[-\left(\frac{x^*}{2\sqrt{\tau_e^*}}\right)^2\right]}{erf\left(-\frac{z}{2L\sqrt{\tau_e^*}}\right)} \tag{31}$$

Evaluating (30) for $x^* = 0$ and $\tau_e^* > 0$ gives

$$\left.\frac{\delta t_e^*}{\delta x^*}\right|_{\substack{x^*=0 \\ \tau_e^*>0}} = \frac{-\ 2/\sqrt{\pi}}{erf\left(-\frac{Z}{2L\sqrt{\tau_e^*}}\right)} \tag{31'}$$

Equation (25) involved in (30) yields for the surface :

$$\left.\frac{\delta t^*_e}{\delta x^*}\right|_{\substack{x^*=0 \\ \tau_e^*>0}} = \frac{-2/\sqrt{\pi}}{erf\left(-\frac{Z, b}{2\,\triangle L\,\sqrt{\alpha_e}}\right)} \tag{32}$$

Solution of Equation (4')

Equation (4') is solved by combining with Equation (24) and Equation (30) simultaneously to give

$$\frac{2\sqrt{\pi}\ exp\left[-(C_2 b)^2\right]}{erf\left(\frac{L}{2L\sqrt{\tau_d^*}}+C_2 b\right)-erf(C_2 b)}=\frac{1}{\lambda_d}\frac{(t_r-t_s)}{(t_s-t_i)}$$

$$\left[L.h_r-\lambda\left\{\frac{-2/\sqrt{\pi}}{erf\quad-\frac{Z}{2L\sqrt{\tau_o^*}}}\right\}\right]\cdots(4'')$$

From equations (25) and (32) the same equation is :

$$\frac{2/\sqrt{\pi}\ exp\left[-(C_2 b)^2\right]}{erf\left[b\left(\frac{1}{2\sqrt{\alpha_d}}+C_2\right)\right]-erf(C_2 b)}=\frac{1}{\lambda_d}\cdot\frac{(t_r-t_s)}{(t_s-t_i)}\Big[L\,h_r$$

$$-\lambda_o\cdot\frac{2/\sqrt{\pi}}{erf\left(\frac{Z.b}{2\triangle L\sqrt{\alpha_o}}\right)}\Big]\quad(4''')$$

Solution of Equation (6")

Substituting equations (25) and (27) into Eq. (6") gives

$$\lambda_d\,(t_s-t_i)\frac{2/\sqrt{\pi}\ exp\left[-\left(\frac{\triangle L}{2L\sqrt{\tau_d^*}}\right)+(C_2 b)^2\right]}{erf\left(\frac{\triangle L}{2L\sqrt{\tau_d^*}}+C_2 b\right)-erf\,(C_2 b)}$$

$$-\lambda_f\,(t_w-t_i)\frac{2/\sqrt{\pi}\ exp\left[-\left(\frac{\triangle L}{2L\sqrt{\tau_f^*}}\right)^2\right]}{erf\left(\frac{1}{2\sqrt{\tau_f^*}}\right)-erf\left(\frac{\triangle L}{2L\sqrt{\tau_f^*}}\right)}$$

$$=\frac{\triangle H_s}{C_p}\ \frac{C_2\ b}{\sqrt{\tau_d^*}}\cdot\lambda_d\qquad(6''')$$

Using equations (25) and (27) yields :

$$\lambda_d\,(t_s-t_i)\frac{2/\pi\ \ exp\ \left[-\left(\frac{b}{2\sqrt{\alpha_d}}\right)+(C_2\ b)^2\right]}{erf\left(\frac{b}{2\sqrt{\alpha_d}}+C_2\ b\right)-erf\,(C_2\ b)}$$

$$-\lambda_f\ (t_w-t_i)\frac{2/\sqrt{\pi}\ \ exp\left[-\left(\frac{b}{2\sqrt{\alpha_f}}\right)^2\right]}{erf\left(\frac{b.L}{2\triangle L\sqrt{\alpha_f}}\right)-erf\left(\frac{b}{2\sqrt{\alpha_f}}\right)}$$

$$= \frac{\Delta H_s}{C_p} \cdot \frac{b^2 \; C_2}{\Delta L \sqrt{\alpha_d}} \cdot \lambda_d \qquad (6^{iv})$$

The last equation expresses the influence of heating through the frozen layer.

Example with Numerical Set of Data : Calculation of 'B'

From (4''') '*b*' can be solved as

$$b = \frac{1}{C_2} \sqrt{1_g \frac{C}{AB}}$$

$$\text{where } A = (t_r - t_s) \left[L.\, h_r + \lambda_e \frac{2/\sqrt{\pi}}{erf\left(\dfrac{Z.\, b}{2 \Delta L \sqrt{\alpha_e}}\right)} \right]$$

$$B = erf \left[b \left(\frac{1}{2\sqrt{\alpha_d}} + C_2 \right) \right] - erf\,(C_2 .\, b)$$

$$C = (t_s - t_i) \cdot \frac{2}{\sqrt{\pi}} \lambda_d$$

For a chosen set of data given by Kan and de Winter in reference[5] the numerical value of '*b*' can be found transcendentally from Eq. (4'''). Assuming that thermoconductivity and thermodiffusivity do not vary much with the pressure and the temperature, the data are :

$P = 0{,}4$ torr, or $P = 4$ torr $t_r - t_s = 40°C$
$t_r = 85°C$, $t_r = 105°C$ $t_s - t_i = 60°C$
$t_s = 40°C$, $t_s = 65°C$ $t_w - t_i = t_r - t_i = 106°C$
$t_i = -21°C$, $t_i = -1°C$
$t_e = 0°C$,
$L = 1{,}5$ cm,
$Z = 1{,}5$ cm,
$\rho_i = \rho_f = 0{,}92$ 9/cm³,
$\rho_d = 0{,}4$ 9/cm³,
$\rho_e = 0{,}29$ 9/cm³,
$\Delta H_s = 670$ cal/g,
$C\rho_f = 0{,}46$ cal/g, K at 25°C, $C\rho_d = 0{,}40$, $C\rho_e = 0.38 \dfrac{cal}{g, K}$
$\sigma = 0{,}7$,
$\lambda_d = 8{,}210^{-5}$ cal/cm, sec, °C,
$\lambda_e = 5{,}62.10^{-6}$ cal/cm, sec, °C,
$\lambda_f = 8{,}2{,}10^{-3}$ cal/cm, sec, °C,
$\Delta P = 0{,}161$ torr,
$hr = 2{,}3.10^{-4}$ cal/cm², sec, °C,
$\varepsilon_r = 0{,}9$,
$\varepsilon_s = 0{,}8$,

Results and Discussion

Now b is found to be 1.10^{-2} (cm. sec$^{-\frac{1}{2}}$). This value has to satisfy all equations.

Equation (10) solved for the boundary of $\triangle L = L$ gives the duration of the process.

$$\tau_{\text{end}} = \left(\frac{\triangle L}{b} \right)^2 = \left(\frac{1,5}{10^{-2}} \right) = 2{,}25.10^4 = 22500 \, [\, sec \,] = 5{,}23 \, [\, hr \,].$$

The result is in good agreement with Kan's results and shows an improvement on the duration.

The interface position versus time (Fig. 2) curve appears similar to the one in reference[2]. The movement of the ice front $\frac{d(\triangle L)}{d\tau}$ and the rate of drying $\frac{d(\triangle L)}{d}$. $\rho_i = N_w$ versus the time are also shown in Fig. 2.

The moisture content change is expressed in the same figure as $(1 - X)$ as well as X versus the time. The plot compares well with the cases given by Gunn and Sandal[7] under similar conditions.

The temperature fields are described by plots of t_d^*, t_e^* and t_f^* versus the distance, for given intervals of the dimensionless times τ_d^*, τ_e^* and τ_f^* (Figs. 3, 4 and 5).

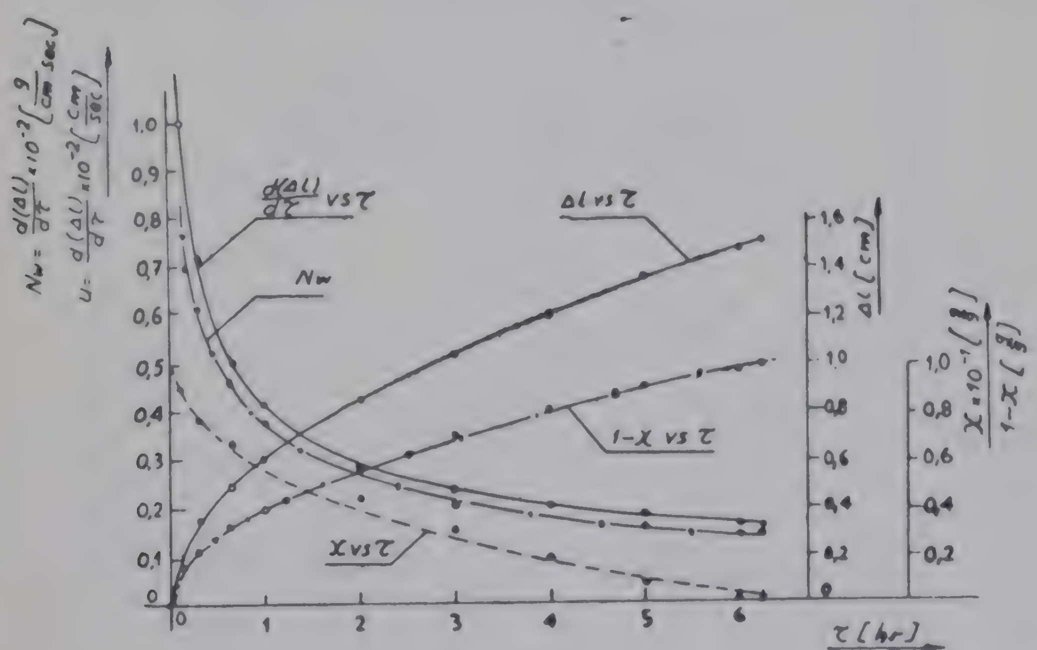
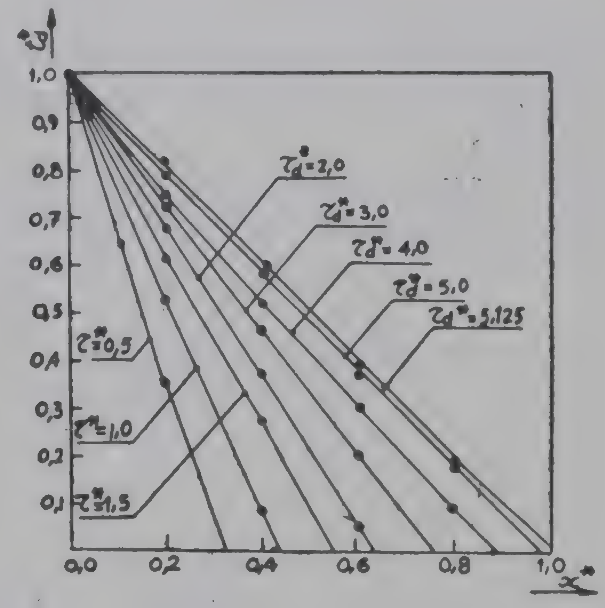
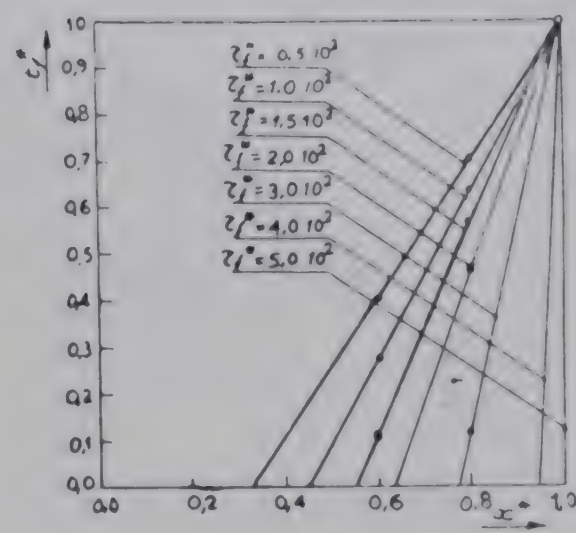


fig2



ϑ^*
1,0
0,9
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,1
0,0
0,2
0,4
0,6
0,8
1,0
x^*
$\tau_d^* = 2,0$
$\tau_d^* = 3,0$
$\tau_d^* = 4,0$
$\tau_d^* = 5,0$
$\tau_d^* = 5,125$
$\tau^* = 0,5$
$\tau^* = 1,0$
$\tau^* = 1,5$

fig 3



$\tau_f^* = 0.5\ 10^3$
$\tau_f^* = 1.0\ 10^3$
$\tau_f^* = 1.5\ 10^3$
$\tau_f^* = 2.0\ 10^2$
$\tau_f^* = 3.0\ 10^2$
$\tau_f^* = 4.0\ 10^2$
$\tau_f^* = 5.0\ 10^2$
c_f^*
x
0,0
0,2
0,4
0,6
0,8
1,0
0,1
0,3
0,5
0,7
0,9

fig 4

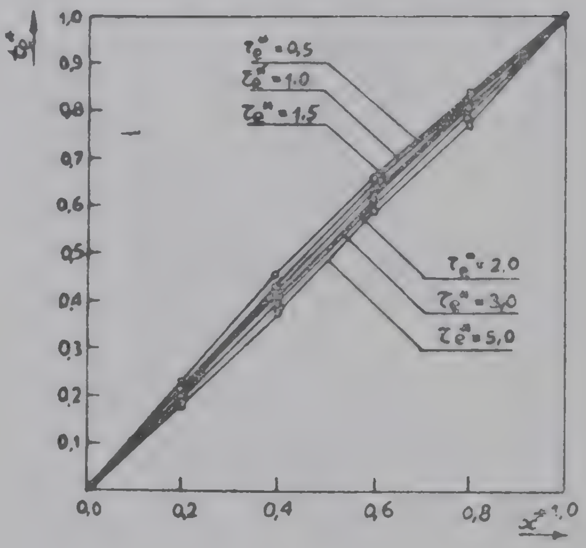


fig. 5.

Nomenclature

t, T	Temperature — °C, °K	
P	Total pressure — torr	
P	Partial pressure — torr	
x	Coordinate in distance of heat fluxes — cm	
q	heat flux for unit area — $\frac{\text{cal}}{\text{cm}^2\ \text{sec}}$	
N_w	Mass flux of the water vapour — $\frac{g}{\text{cm}^2\ \text{sec}}$	
τ	time — sec	
λ	Thermal conductivity (effective value) cal / (cm) (sec) (°C)	
α	Thermal diffusivity (effective value) $\frac{\text{cm}^2}{\text{sec}}$	
h	heat transfer coefficient — $\frac{\text{cal}}{\text{cm}^2\ \text{sec}\ °\text{C}}$	
ε	Emissivity — dimensionless	
σ	Porosity, Stephan Boltzmah's constant in Eq. (5) — dimensionless	
$\triangle H$	Latent heat of sublimation — cal / g	
L	Thickness of the specimen — cm	
$\triangle L$	Thickness of the dried region, synonymous with the position of the ice front — cm	
u	Velocity of movement of the ice front inward the specimen — $\frac{\text{cm}}{\text{sec}}$	
C_p	Specific heat (effective value) — $\frac{\text{cal}}{\text{g}°\text{C}}$	
X	Fraction of initial moisture remaining in the specimen $\frac{\text{g}}{\text{g}}$	
ρ	Density (effective value) — $\frac{\text{g mass}}{\text{cm}^3}$	
b	Coefficient from Eq. (10) — $\frac{\text{cm}}{\text{sec}^{\frac{1}{2}}}$	
η	Complex, defined in Eq. (18)	
U	Complex, defined in Eq. (19)	
z	distance, between the radiator and the surface of the specimen — cm.	

Subscripts

i	ice front, ice
c	at the condenser surface

s	at the surface of the specimen, sublimation
r	at the radiator
d	dried region
e	for the space between the surface and the radiator
f	for the frozen region
w	at the wall, water vapour
*	dimensionless.

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Design and Performance of the Pulse Tube

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Pulse tube is a miniature cryorefrigerator, invented by Gifford and his associates around 1960. It uses helium or air as the working substance and produces very low temperatures even with moderate pressure ratios.

This paper reports the theoretical relations which have been developed using the concept of steps. These relations express the temperature ratio under zero heat pumping condition (θ_o), the heat pumping rate (Q_o) and refrigeration due to Joule-Thomson effect (Q_J) in terms of parameters of the tube and the working substance under certain assumptions. One of the important assumptions made in the present investigation is that the gas portion which never leaves the hot end volume undergoes isothermal compression and expansion. The relation for θ_o shows that θ_o varies slightly due to variation in pressure ratio (γ_p) These relations for θ_o predict values of cold end temperatures slightly higher than those predicted by using Longsworth's relation.

The gas admitted into the pulse tube performs work on the inside gas and thus produce refrigeration. Relation for heat pumping rate indicates that Q_o is directly proportional to the high pressure and it decreases linearly with $\theta^{1/n}$ (where θ is temperature ratio and n is number of steps). Helium gives larger Q_o compared to air for the same geometric and operating conditions.

Under the given set of conditions, the gross refrigeration (the heat pumping rate plus the refrigeration due to Joule-Thomson effect) produced by air is less than that produced by helium for values of high pressure (P_H) upto about 10 kgf/sq cm abs. However, for higher values of P_H, the gross refrigeration produced by air is more than that produced by helium.

With these relations, a design procedure for a pulse tube is outlined.

The analysis is further extended to multistage units to obtain optimum working conditions.

Pressure Drop During Flow Boiling of a Mixture of Refrigerants Freon–12 and Freon–22

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Homogeneous flow theory provides the simplest technique for analysing single component two-phase flow. Total pressure drop data are predicted for various values of exit qualities. This technique is extended to flow boiling of a non-azeotropic binary mixture of refrigerants R–12 and R–22. Total pressure drop data are predicted for various percentage composition values with varying quality of each of the components.

Introduction

Extensive use of horizontal tube evaporators by the refrigeration and air-conditioning industry and their role as a major cost and performance item requires that they be designed from an accurate knowledge of heat transfer and flow characteristics.

The use of a single component as a refrigerant is confined to isothermal refrigeration. But in practice, non-isothermal refrigeration would be desirable in several systems as for example in process refrigeration duty, where products are to be cooled through a range of temperatures. A non-azeotropic mixture of refrigerants with non-isothermal change of phase would reduce the amount of irreversibility considerably as in Lorentz cycle and make it ideally suited for such a system. It is possible to achieve optimum working conditions for a single stage compressor, with a wide range of refrigerating capacities and temperatures in the evaporator, by suitably controlling the composition of the working mixture. Power characteristics can be improved and the list of working substances could be extended thus increasing the possibilities for unifying the equipment.

Forced convection evaporation is one of the most complex and the least understood heat transfer process. Satisfactory prediction of performance is still not possible even after a tremendous amount of research effort in this direction. Most of the difficulty stems from the fact that the flow patterns, heat transfer rate and pressure drop are markedly affected by a large number of variables. As the fluid progresses downstream from the entrance of an evaporator tube, the flowing quality

increases and the flow may pass through any or all of the flow regimes; bubbly flow, bubbly slug flow, slug flow, churn flow, wavy annular flow until finally annular flow is established with increase in flow velocity resulting in suppression of nucleation.[1*]

In refrigeration work, the fluid usually enters the evaporator through a throttle valve so that there is no sub-cooled region and if dry-out is not reached, the evaporator operates in the region of bulk boiling with the flow pattern predominantly annular.

Various analytical models[2] used in analysing two-phase flow are: homogeneous flow, separated flow and drift-flux model. The most extensively used in predicting pressure drop is the one developed by Martinelli Nelson[3] for boiling water using separated flow model. Another widely used correlation is due to Pierre[4]. It has been observed that Pierre's correlation underestimates pressure drop data whereas Martinelli Nelson's overestimates, with partial evaporation. The tendency for Martinelli Nelson's correlation to predict conservative results for refrigerants within 10 to 30 per cent has been observed by several investigators[5–8].

In most of the investigations Martinelli Nelson's correlation is extended to other fluids in the same Prandtl number range. This method involves prediction of a relation between void fraction and vapour quality.

The choice of an analytical model for the description of the two-phase flow is really complicated for a single component itself and the problem becomes all the more complex for the flow boiling of a binary, non-azeotropic mixture of refrigerants. As a first step to solve this problem, the homogeneous flow technique, the simplest of all, may be used to predict pressure drop data. In this paper, total pressure drop data are predicted for single component refrigerants R-12 and R-22 and a non-azeotropic mixture of R-12 and R-22, by using homogeneous flow model.

Analysis

In the homogeneous flow model, the average values of velocity, temperature and chemical potential for each phase are same and the mixture is treated as a pseudo fluid that obeys the usual equations of single component flow. The pseudo properties are the weighted averages and are not necessarily the same as properties of either phase.

From the momentum equation for one dimensional steady homogeneous equilibrium flow, total pressure gradient is given by

$$\left(\frac{dp}{dz}\right)_T = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_A + \left(\frac{dp}{dz}\right)_G \tag{1}$$

For a boiling fluid, flowing through a horizontal duct of constant area with uniform heat flux, neglecting flashing and compressibility, the total pressure gradient will be simplified as:

$$\left(\frac{dp}{dz}\right)_T = \frac{2\,C_F \mathbf{G}^2}{D}\,(V_f + xV_{fg}) + \mathbf{G}^2 V_{fg}\,\frac{dx}{dz} \tag{2}$$

Evaluation of Friction Pressure Gradient

1. Two-phase Flow of a Single Component

Single phase friction factors in turbulent flow are usually correlated in terms of Reynolds number and pipe roughness, whereas Meyer and Wallis[9] recommended the use of a constant value of 0.005 as two-phase turbulent friction factor, for high velocity annular mist flow regime. For flows with either very low quality or very high quality, the more plentiful phase plays a dominant role in determining the mixture properties and the friction factor.

McAdams *et al.*[10] and Owens[11] proposed a method of evaluating friction factor from some equivalent single phase flow. For low quality vapour-liquid mixtures, it may be assumed that the friction factor is the same as it would be if the total mass flowed entirely as liquid.

Two-phase multiplier for friction pressure gradient is now defined as follows :

$\phi^2_{fo} = (dp / dz)_F \; / \; (dp / dz)_{Ffo}$

This can be simplified as

$$\phi^2_{fo} = 1 + x\left(\frac{\rho f}{\rho g} - 1\right) \tag{3}$$

Hence friction pressure gradient for two-phase flow is :

$$\left(\frac{dp}{dz}\right)_F = \left[1 + x\left(\frac{\rho f}{\rho g} - 1\right)\right] \frac{2\, C_{Ffo}\, G U_{fo}}{D} \tag{4}$$

On the same lines, for high quality two-phase flow, friction factor and other properties are determined from vapour phase. Friction pressure gradient in this case is :

$$\left(\frac{dp}{dz}\right)_F = \left[\frac{\rho g}{\rho f} + x\left(1 - \frac{\rho g}{\rho f}\right)\right] \frac{2 \cdot C_{Fgo}\, G\, U_{go}}{D} \tag{5}$$

where two-phase multiplier ϕ^2_{go} is given by

$$\phi^2_{go} = \frac{\rho g}{\rho f} + x\left(1 - \frac{\rho g}{\rho f}\right) \tag{6}$$

When the quality is neither small nor very large, so that neither phase dominates in determining the properties, one of the expressions for equivalent viscosity is used in the Reynolds number and single phase friction factor charts may be used.

Using Blasius equation for smooth pipe flow

$$C_F = 0.079\, (\mathrm{Re})^{-0.25} \tag{7}$$

and McAdam's[10] relation for equivalent viscosity, two-phase multiplier may be expressed as :

$$\phi^2_{fo} = \left[1 + x\left(\frac{\rho f}{\rho g} - 1\right)\right]\left[1 + x\left(\frac{\mu f}{\mu g} - 1\right)\right]^{-0.25} \tag{8}$$

Similarly it may be shown with reference to vapour

$$\phi^2_{go} = \left[\frac{\rho g}{\rho f} + x\left(1 - \frac{\rho g}{\rho f}\right)\right]\left[\frac{\mu g}{\mu f} + x\left(1 - \frac{\mu g}{\mu f}\right)\right]^{-0.25} \tag{9}$$

2. Two-phase Flow of a Binary Mixture of Refrigerants

Let K be the percentage of component 1 in a mixture of 1 and 2. Extending homogeneous flow technique to a mixture, weak in 1 and rich in 2, the properties are determined by component 2.

Hence $C_{Fm} = C_{fo_2}$ and $\mu_m = \mu_2$

A two-component multiplier is now defined as

$$\phi^2_{0_2} = \frac{(dp/dz)_{Fm}}{(dp/dz)_{Fo_2}}$$

on evaluation of this equation one obtains

$$\phi^2_{02} = \frac{\rho_2}{\rho_m}$$

Since $\rho_m = K\rho_1 + (1-k)\rho_2$, the following equation is obtained

$$\phi^2_{0_2} = \frac{\rho_2}{K\rho_1+(1-k)\rho_2} \tag{10}$$

Thus friction pressure gradient for the mixture is

$$\left(\frac{dp}{dz}\right)_{Fm} = \phi^2_{0_2}\left(\frac{dp}{dz}\right)_{Fo2} \tag{11}$$

Now $\left(\frac{dp}{dz}\right)_{Fo_2}$ for component 2 may be evaluated using either equation (8) or (9) depending upon its quality. Similarly for a mixture rich in component 1, two-component multiplier and friction pressure gradient may be expressed as

$$\phi^2_{01} = \frac{\rho_1}{K\rho_1+(1-k)\rho_2} \tag{12}$$

and

$$\left(\frac{dp}{dz}\right)_{Fm} = \phi^2_{01}\left(\frac{dp}{dz}\right)_{F01} \tag{13}$$

where $\left(\frac{dp}{dz}\right)_{F01}$ may be evaluated using either equation (8) or (9) depending upon its exit quality.

In case of a mixture, where neither component is dominating, two-component multiplier may be evaluated as

$$\phi^2_{02} = \frac{\left(\frac{dp}{dz}\right)_{Fm}}{\left(\frac{dp}{dz}\right)_{Fo_2}}$$

on evaluation of this equation, one obtains

$$\phi^2_{0_2} = \frac{U_m}{U_{0_2}}\left(\frac{Re}{Re_2}\right)^{-0.25}$$

The following equation is obtained with further simplification

$$\phi^2_{o2} = \frac{\rho_2}{\rho_m}\left(\frac{\mu_m}{\mu_2}\right)^{0.25} \tag{14}$$

Cicchitti's[12] expression for equivalent viscosity is now used.
Thus

$$\mu_m = K\mu_1 + (1-k)\mu_2 \tag{15}$$

Combining equation (15) with (14), we get

$$\phi^2_{o2} = \left[\frac{\rho_2}{K\rho_1+(1-k)\rho_2}\right] \left\{\frac{1}{\mu_2}\left[K\mu_1 + (1-k)\mu_2\right]\right\}^{0.25} \tag{16}$$

Similarly it may be shown that two-component multiplier with reference to component 1 is given by

$$\phi^2_{o1} = \left[\frac{\rho_1}{K\rho_1 + (1-k)\rho_2}\right] \left[\frac{K\mu_1 + (1-k)\mu^2_2}{\mu^1_1}\right]^{0.25} \tag{17}$$

Friction pressure gradient for binary mixture may be obtained from either equation (11) or (13).

Evaluation of Acceleration Pressure Gradient

1. Two-phase Flow of a Single Component

For single component, from momentum equation, acceleration pressure gradient is obtained as

$$\left(\frac{dp}{dz}\right)_A = G^2\frac{d}{dz}\left(\frac{1}{\rho}\right) - \frac{G^2}{\rho A}\ \frac{dA}{dz}$$

This can be re-written in terms of quality. Thus

$$\left(\frac{dp}{dz}\right)_A = G^2\frac{d}{dz}\left(\frac{x}{\rho_g} + \frac{1-x}{\rho_f}\right) - \frac{G^2}{\rho.A}\ \frac{dA}{dz}$$

on further simplification, the following equation results :

$$\left(\frac{dp}{dz}\right)_A = G^2\left\{ V_{fg}\frac{dx}{dz} + \frac{dp}{dz}\left[x\frac{dV_g}{dp} + (1-x)\frac{dV_f}{dp}\right] - (V_f + x V_{fg})\ \frac{1}{A}\ \frac{dA}{dz}\right\} \tag{18}$$

Neglecting compressibility and considering flow through duct of constant area, equation (18) can be written as :

$$\left(\frac{dp}{dz}\right)_A = G^2\ V_{fg}\ \frac{dx}{dz} \tag{19}$$

2. Two-phase of a Binary Mixture

For a binary mixture with reference to component 2, we can write

$$\left(\frac{dp}{dz}\right)_A = G^2\ V_{fg}{}^2\ \frac{dx_2}{dz} \tag{20}$$

whereas with reference to component 1, it is expressed as

$$\left(\frac{dp}{dz}\right)_A = G^2 \; V_{fg}{}^1 \; \frac{dx_1}{dz} \tag{21}$$

Total Pressure Drop

1. Single Component Flow

Total pressure gradient is obtained by

$$\left(\frac{dp}{dz}\right)_T = \frac{2C_{Pfo}\, G^2}{D\, \rho_f}\, \phi^2{}_{fo} + G^2\, V_{fg}\, \frac{dx}{dz} \tag{22}$$

From energy equation with constant heat flux, negligible work and negligible kinetic or potential energy changes, it can be written as :

$$\frac{dx}{dz} = -\frac{4\, q}{G\, Dh_{fg}} \tag{23}$$

which shows that quality of vapour increases linearly with distance 'z' along the duct. Letting quality $x = 0$ at $Z = 0$, equation (22) becomes after integration and non-dimensionalising,

$$\frac{2\, \rho_f\, (\Delta\, P)_T}{G^2} = 0.316\, (Re_{fo})^{-0.25}\, \frac{Z}{D}\, \phi^{-2}_{fo} + 2\left(\frac{\rho_f - \rho_g}{\rho_g}\right) x \tag{24}$$

where $$\phi^{-2}_{fo} = \frac{1}{x}\int_o^x \phi^2_{fo}\, dx \tag{25}$$

The non-dimensional pressure drop as expressed in terms of the properties of vapour will be as follows :

$$\frac{2\, \rho_g\, (\Delta\, P)_T}{G^2} = 0.316\, (Re_{go})^{-0.25}\, \frac{Z}{D}\, \phi^{-2}_{go} + 2\left(\frac{\rho_f - \rho_g}{\rho_f}\right) x \tag{26}$$

where $$\phi^{-2}_{go} = \frac{1}{x}\int_o^x \phi^2_{go}\, dx \tag{27}$$

2. Binary Mixture Flow

Using equations (11), (16) and (20) total pressure gradient for binary mixture may be expressed as

$$\left(\frac{dp}{dz}\right)_{T_m} = 0.158\; \phi^2_{02}\; \phi^2_{f02}\; (Re_{fo2})^{-0.25}\; \frac{G_2^2}{{}_{f_2} D} + G_2^2\left(\frac{\rho f^2 - \rho g^2}{\rho g^2 - \rho f^2}\right) x_2 \tag{28}$$

Integration of equation (28) yields an expression for pressure drop. The resulting expression after non-dimensionalisation is as follows :

$$\frac{2 Pf_2(\Delta P)\, T_m}{G_2^2} = 0.316\;\; \phi^2_{02}\; \phi^{-2}_{fo2}\, \frac{Z}{D}\, (\, Re_{fo2})^{-0.25} + \left(\frac{Pf_2 - Pg_2}{Pg_2}\right)\; x_2 \tag{29}$$

where $$\phi^{-2}_{fo2} = \frac{1}{x}\int_o^x \phi^2_{fo2}\, dx \tag{30}$$

Similarly pressure drop in terms of vapour properties may be expressed as :

$$\frac{2\rho_{g2}(\Delta P)_{Tm}}{G_2^2} = 0.316\ \phi^2_{02}\ \phi^2_{g02}\ \frac{Z}{D}(Re_{go_2})^{-0.25} + \left(\frac{\rho_{f2}-\rho_{g2}}{\rho_{f2}}\right) \qquad (31)$$

$$\text{where}\ \phi^{-2}_{g02} = \frac{1}{x}\int_0^x \phi^2_{g02}\,dx \qquad (32)$$

Results and Discussion :

Two-phase multipliers for friction pressure gradient ϕ^2_{fo} and friction pressure drop ϕ^{-2}_{f} are evaluated for refrigerants Freon-12 and Freon-22 for the entire pressure range from 0.2 to 40.0 Kg/cm² (critical) with out-let quality of the vapour varying from 0 to 1.0 using equations (8) and (25). The data evaluated are tabulated in Tables 1 and 2 for few values of pressure encountered in refrigeration.

TABLE 1. *Two-phase multipliers ϕ^2_{fo} and ϕ^2_{fo} for Freon-12*

Quality x	Pressure (Kgf/cm²)							
	1.5		2.0		3.0		4.0	
	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}
0.20	20.98	11.19	16.25	8.74	11.25	6.16	8.76	4.89
0.40	35.42	19.52	27.36	15.14	18.81	10.51	14.56	8.20
0.60	48.09	26.81	37.11	20.74	25.46	14.32	19.66	11.12
0.80	59.72	33.41	46.06	25.88	31.56	17.81	24.35	13.81
1.00	70.64	39.75	54.47	30.70	37.30	21.10	28.75	16.32

TABLE 2. *Two-phase multipliers ϕ^2_{fo} and ϕ^2_{fo} for Freon-22*

Quality x	Pressure (Kgf/cms)							
	2.0		3.0		5.0		7.0	
	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}	ϕ^2_{fo}	ϕ^{-2}_{fo}
0.20	20.26	10.79	14.09	7.61	8.85	4.93	6.58	3.77
0.40	34.25	18.85	23.70	13.14	14.73	8.28	10.82	6.18
0.60	46.52	25.91	32.14	17.98	19.89	11.24	14.56	8.31
0.80	57.79	32.38	39.90	22.43	24.65	13.96	18.00	10.28
1.00	68.37	38.45	47.18	26.60	29.11	16.52	21.23	12.12

The total pressure drop data are computed for the above refrigerants. Fig. 1 shows the comparison of the values of total pressure drop predicted in this investigation with the experimental values of Charles Johnston[13]. The predicted pressure drop data agree with the measured data of Charles Johnston within ± 25per cent.

Similarly for a binary mixture of Freon-12 and Freon 22, two-component multipliers, overall multiplying factors for friction pressure gradient (product of two-

component and two-phase multiplier) ϕ^2. ϕ^2_{fo}) and for friction pressure drop (product of ϕ^2 and ϕ^{-2}_{fo}) have been computed for various values of percentage composition K and outlet quality of each of the components. The data of overall multiplying factors evaluated for few values of pressure are shown in Table 3. With the data covering

TABLE 3. *Overall Multiplying Factors for Binary Mixture of R-12 and R-22*

Percentage composition K	Quality x2	Quality x1	1.5 Kgf/cm2		2.0 Kgf/cm2		3.0 Kgf/cm2		5.0 Kgf/cm2		7.0 Kgf/cm2	
			Φ^2	Φ^{-2}	Φ^2	Φ^{-2}	Φ^2	Φ^{-2}	Φ^2	Φ^{-2}	Φ^2	Φ^{-2}
		0.4	39.64	22.29	30.83	17.37	21.21	12.01	13.35	7.68	9.6	5.61
	0.4	0.6	45.14	25.38	35.13	19.79	24.17	13.69	15.15	8.71	10.92	6.38
		0.8	48.62	27.34	37.86	21.33	26.06	14.76	16.30	9.37	11.77	6.88
		0.4	45.85	25.96	35.62	20.20	24.45	13.92	15.41	8.86	11.02	6.40
0.4	0.6	0.6	53.86	30.50	41.86	23.73	28.73	16.35	18.01	10.36	12.89	7.49
		0.8	59.24	33.54	46.06	26.12	31.63	18.01	19.78	11.37	14.18	8.25
		0.4	49.89	28.33	38.73	22.02	26.55	15.14	16.76	9.63	11.94	6.40
	0.8	0.6	59.88	34.01	46.50	26.44	31.88	18.18	20.01	11.50	14.26	7.49
		0.8	66.91	37.99	51.98	29.55	35.65	20.33	22.30	12.82	15.92	8.25

entire pressure range, pressure drop at any distance Z along the evaporator coil may be evaluated for rapid design predictions.

Nomenclature

A	=	Area of cross section of pipe (m^2)
C_F	=	Friction factor
D	=	Diameter of pipe (m)
$\frac{dp}{dz}$	=	Pressure Gradient (Kgf/m^2 per metre length)
G	=	Mass flux (Kg/sec.m^2)
h_{fg}	=	Latent heat of evaporation (Kcal/kg)
m	=	Mass flux (Kg/sec. m^2)
q	=	Heat flux (Kcal/sec. m^2)
Re	=	Reynolds Number
U	=	Velocity (m/sec.)
V	=	Specific volume (m^3/kg)
V_{fg}	=	$V_g - V_f$
x	=	Quality of vapour
z	=	Co-ordinate in direction of motion (m)
Δp	=	Pressure drop Kgf/m^2
ρ	=	Density of fluid Kg/m^3
μ	=	Absolute viscosity (Kg/sec. m)
ϕ^2	=	Two-component multiplier
ϕ^2_{fo}	=	Two-phase multiplier in terms of equivalent single phase liquid flow.
ϕ^2_{go}	=	Two-phase multiplier in terms of equivalent single phase vapour flow.

ϕ^{2}	=	Overall multiplying factor for friction pressure gradient for mixture $[\phi^2 . \phi_{fo}^{2}]$
ϕ^{-2}	$\equiv$	Overall multiplying factor for friction pressure drop for mixture $[\phi^2 \ \phi_{fo}^{2}]$

Subscripts

A	=	Acceleration
F	=	Friction
f	=	Liquid
fo	=	equivalent liquid flow
fo_1	=	equivalent liquid flow of component 1
fo_2	=	equivalent liquid flow of component 2
G	=	Gravity
go	=	equivalent vapour flow
go_1	=	equivalent vapour flow of component 1
go_2	=	equivalent vapour flow of component 2
m	=	binary mixture of 1 and 2
0_1	=	equivalent flow of component 1
0_2	=	equivalent flow of component 2
T	=	Total

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Experimental Studies on Heat and Mass Transfer During Precooling of Model Foods

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Introduction

Food products are generally precooled *in situ* before they are transported for further processing. This operation is performed in order to remove the field heat or processing heat so that the ripening or deterioration of the product is slowed down.

Experimental investigations on aircooling of foodstuffs are reported[1–7]. Food products are moist porous bodies. Hence, any cooling or heating operation is accompanied by a movement of moisture within the product and into the ambient by evaporation at the surface. The coupled heat and mass transfer effects of this surface evaporation on the overall heat transfer is significant. This aspect has been theoretically analysed by Srinivasa Murthy *et al.*[8]

It is observed from a study of literature that though a number of experimental studies are available on aircooling of food products, most of them are conducted on specific commodities under particular processing conditions. Generalised process studies which deal with the effects of various parameters are rare. In the present work, model food gels are cooled in an 'airconditioning tunnel' at various air temperatures, humidities and velocities. Model foods are constructed out of Agar-Agar, sugar, soluble starch and water. These are similar to the one suggested by Dyner and Hesselschwerdt.[9] The temperature profile and moisture loss characteristics are presented for various processing parameters and their effects are discussed. The results are compared with the theoretical predictions of the authors[8] and a very good agreement is observed between the two. The evaporation of moisture from the product is found to have a significant effect on the heat transfer characteristics. Higher cooling rates accompanied by higher moisture losses are obtained at lower air humidities and higher velocities.

Experimental Investigations

Food Models : In the present work, model foods are used as experimental specimens. Use of food models for process studies has the following advantages :

i. They are homogeneous and have uniform structure.

ii. They can be cast into perfect geometric shapes.

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iii. Thermocouples and other probes can be embedded at exact locations by locating them in the mould before casting.

iv. Product having the same properties and composition can be prepared again and again whereas it is impossible to obtain an actual food product having identical properties.

In the present study, the model food made of Agar-Agar, sugar and water (in proportion of 3 : 30 : 100 by weight) is used. The procedure of preparation of the model is similar to the one discussed by Dyner and Hesselschwerdt[9]. Nickel plated steel moulds which can produce slabs of dimensions 75 mm × 75 mm × 10 mm (or 15 mm) are used. 42 gauge (0.01 mm diameter) copper-constantan thermocouples are held in position by taut threads and the model is cast in it. The specimen is equilibrated to a predetermined initial temperature in a desiccator maintaining 100 per cent humidity and kept in thermostatic bath.

The airconditioning tunnel : The schematic diagram of the 'airconditioning tunnel' in which the precooling experiments are conducted is shown in Fig. 1. The test section has the dimensions 16 cm × 16 cm × 30 cm (length) and sufficient lengths of converging and diverging portions are provided to obtain a smooth and uniform flow. As may be seen in Fig. 1, heating cooling coils are provided at the entrance of the tunnel. Cooled alcohol from a thermostat is circulated in the cooling coil which not only cools the air but also acts as a dehumidifier. By suitably adjusting the heating and cooling intensities, desired temperature and humidity conditions can be achieved at the test section. The air flow velocity can be controlled by operating the throttle valve provided at the exhaust end of the tunnel. The tunnel is well insulated with 1 inch thick thermocole. The pressure drop across the orifice meter placed as shown in the figure gives the value of velocity at the test section. This reading has been previously calibrated with respect to a swinging wane

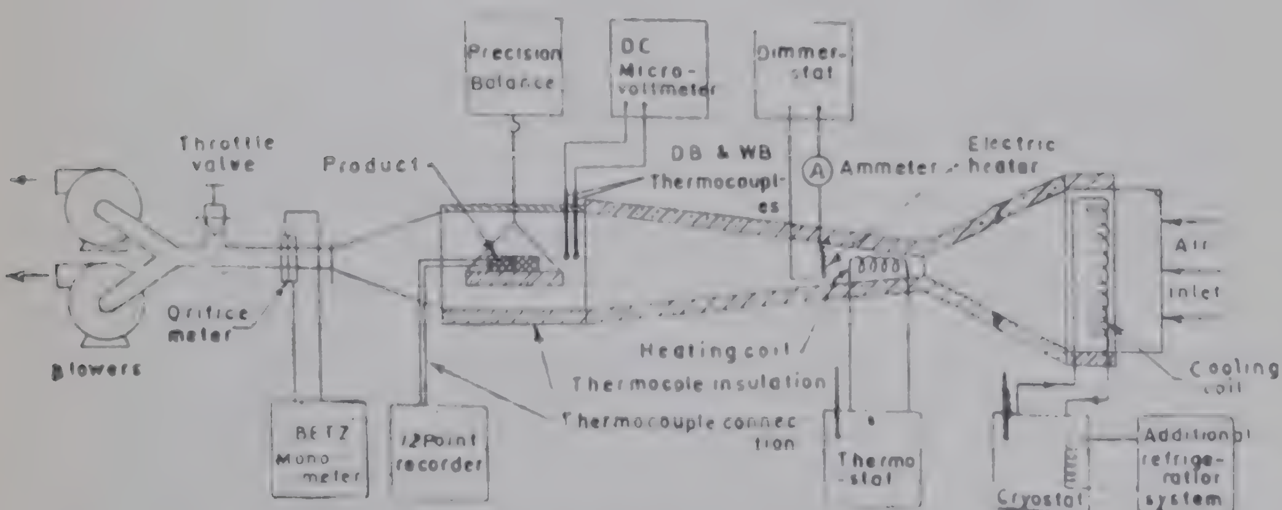

FIG. 1. Schematic diagram of the humidity tunnel

anemometer (accuracy 0.01 m/sec.) placed at centre of the test section. The dry bulb and wet bulb temperatures are measured with two thermocouples (one with a wet cotton wick around its bead) located at the entrance to the test section. Temperature, humidity and velocity traverse are conducted at various stations in the test section for different ambient conditions and are found to be uniform over the centre of the section.

Experimental procedure : The tunnel is put into operation by starting the blowers and setting the required velocity by adjusting the throttle valve. The cold alcohol flow in the cooling coil is restored. After allowing for steady state to be attained, the dry and wet bulb temperatures are measured. If the required conditions are not attained, the heating and cooling intensities are adjusted. For instance, if it is needed to dehumidify further maintaining the same dry bulb temperature, the cooling intensity is increased and the heating is also increased slightly. Required conditions could be achieved in 2 or 3 trials. Then the tunnel is operated for atleast one hour to attain steady conditions. The tunnel, being an open type one, is sensitive to the changes in ambient conditions. Hence, the tests are conducted during late night hours when the fluctuations in the ambient conditions are minimum. After the attainment of steady state, the air flow is stopped, the product is suspended in the test section from a precision balance (0.005 g accuracy), the thermocouple connections are made and the airflow is restored. The whole set of operations take less than 30 seconds. The temperatures are recorded on a 12-channel potentiometer recorder. The weight changes are noted down at intervals.

Results and Discussion

The time-temperature variations during aircooling of the model food gels at various operating conditions are presented in figures 2 to 5. As can be expected, faster cooling rates are observed at lower humidities. However, only the surface temperature drops sharply at lower humidities compared to that in the interior of the product. This effect diminishes as the air humidity increases, thus supporting the fact that the surface evaporation effects are significant. It may be observed that at 70 per cent humidity, the product temperatures have dropped below the dry bulb temperature of the cooling air. The fact that higher air velocities and lower air temperatures hasten the cooling is obvious from the figures 3 and 4. Fig. 5 shows the effect of the initial product temperature. The initial steep cooling curve for the higher product temperature is due to the higher sensible heat exchange at the beginning of the process due to larger temperature differential. Figures 6 and 7 show the temperature profiles at various depths of the product. It is again clear from Fig. 6 that at the low humidity of 66 per cent the product temperature has dropped to a value much lower than that of the ambient. A comparison is made with the theoretical predictions of the authors[8] where the effect of surface evaporation is considered. A good agreement is observed between the theory and the experiments. For the purpose of comparison, the time-temperature histories calculated by conventional methods (like that of Pflug and Kopelman[10] where only pure convection at the surface is considered are also presented. It is seen that the temperatures are over-

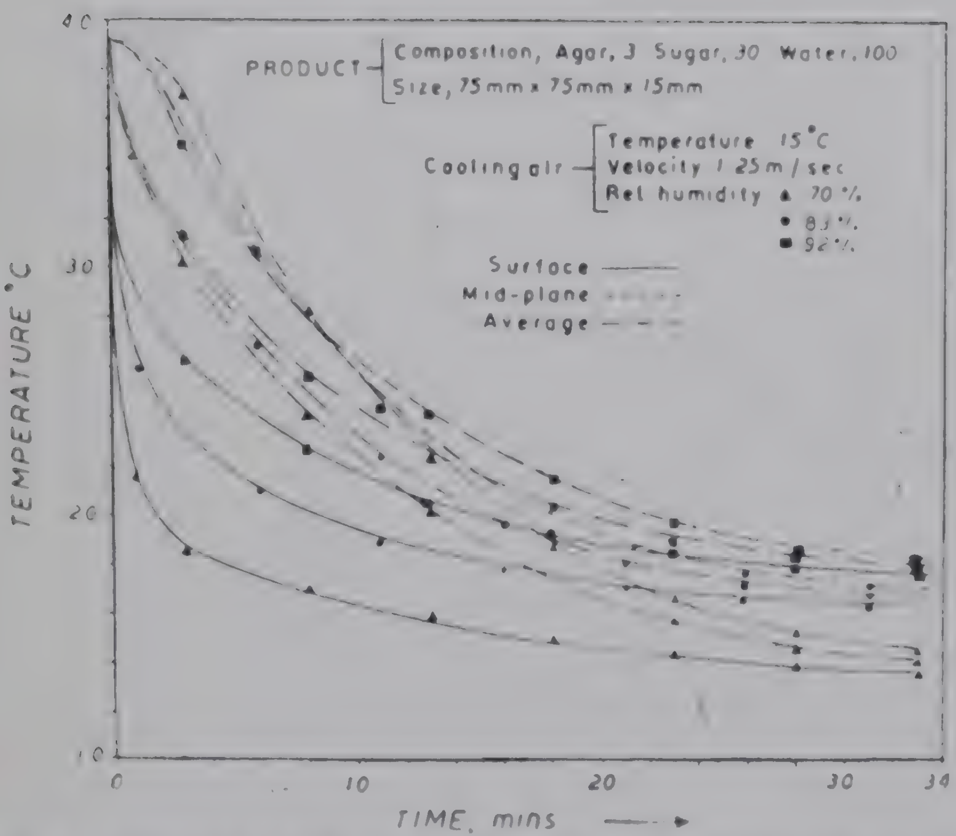

FIG. 2. Aircooling of model food gel effect of moisture content in cooling air

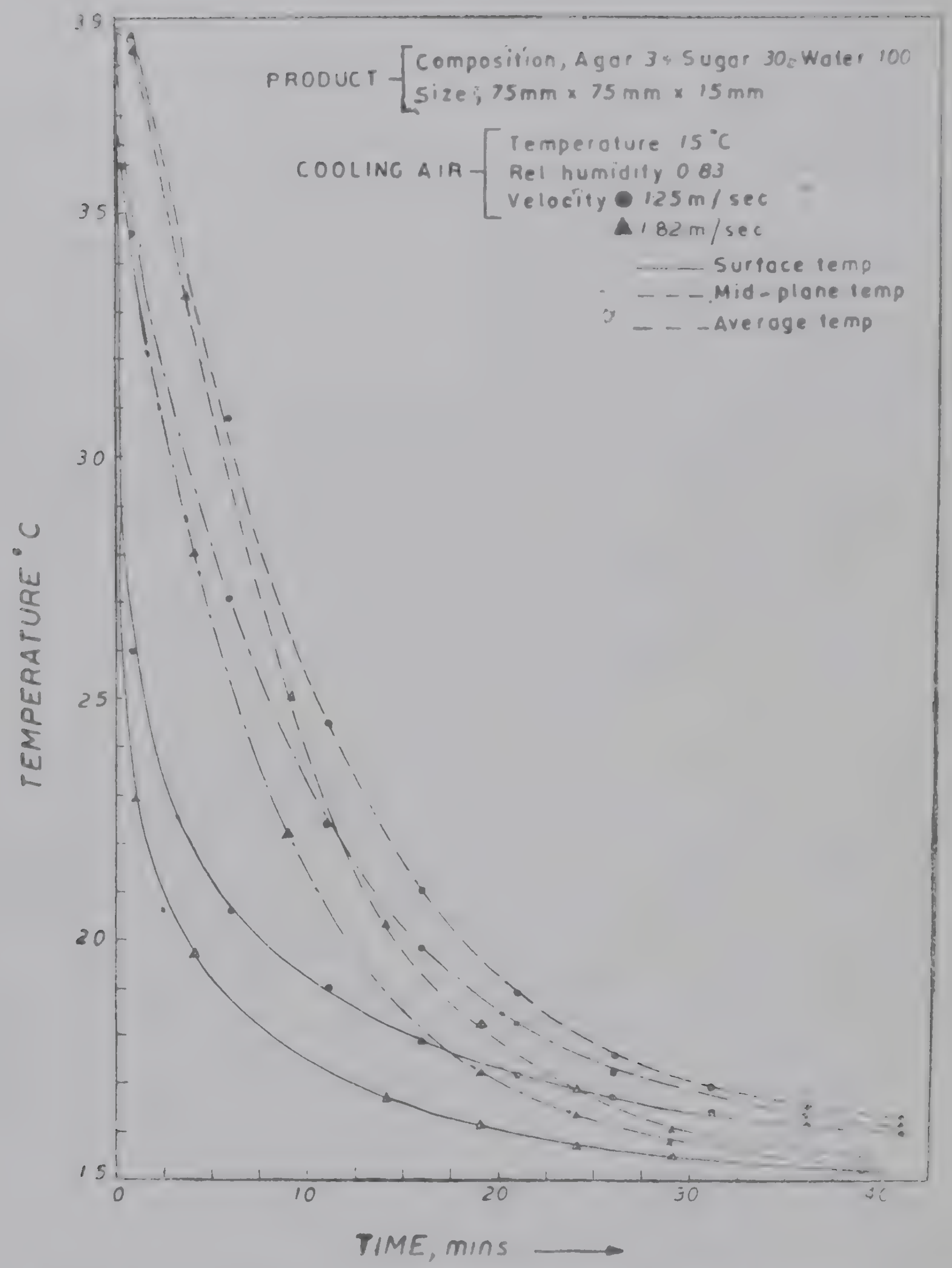

FIG. 3. Aircooling of model food gel — effect of cooling air velocity

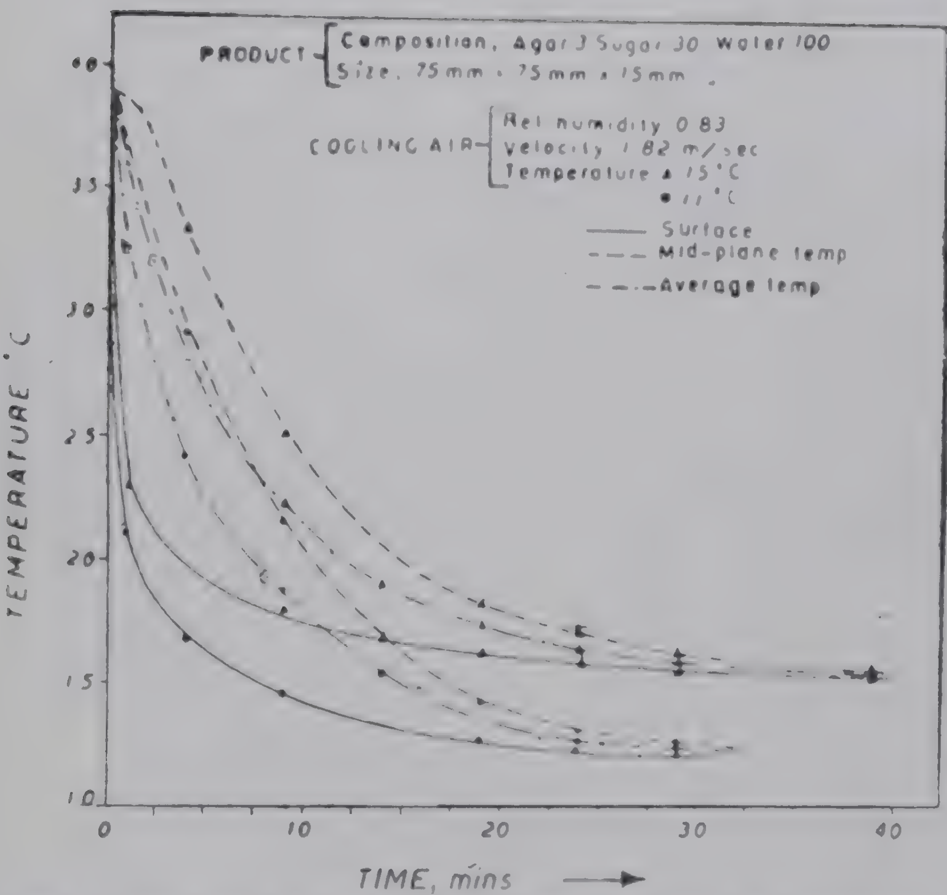

FIG. 4. Aircooling of model food gel — effect of cooling air temperature

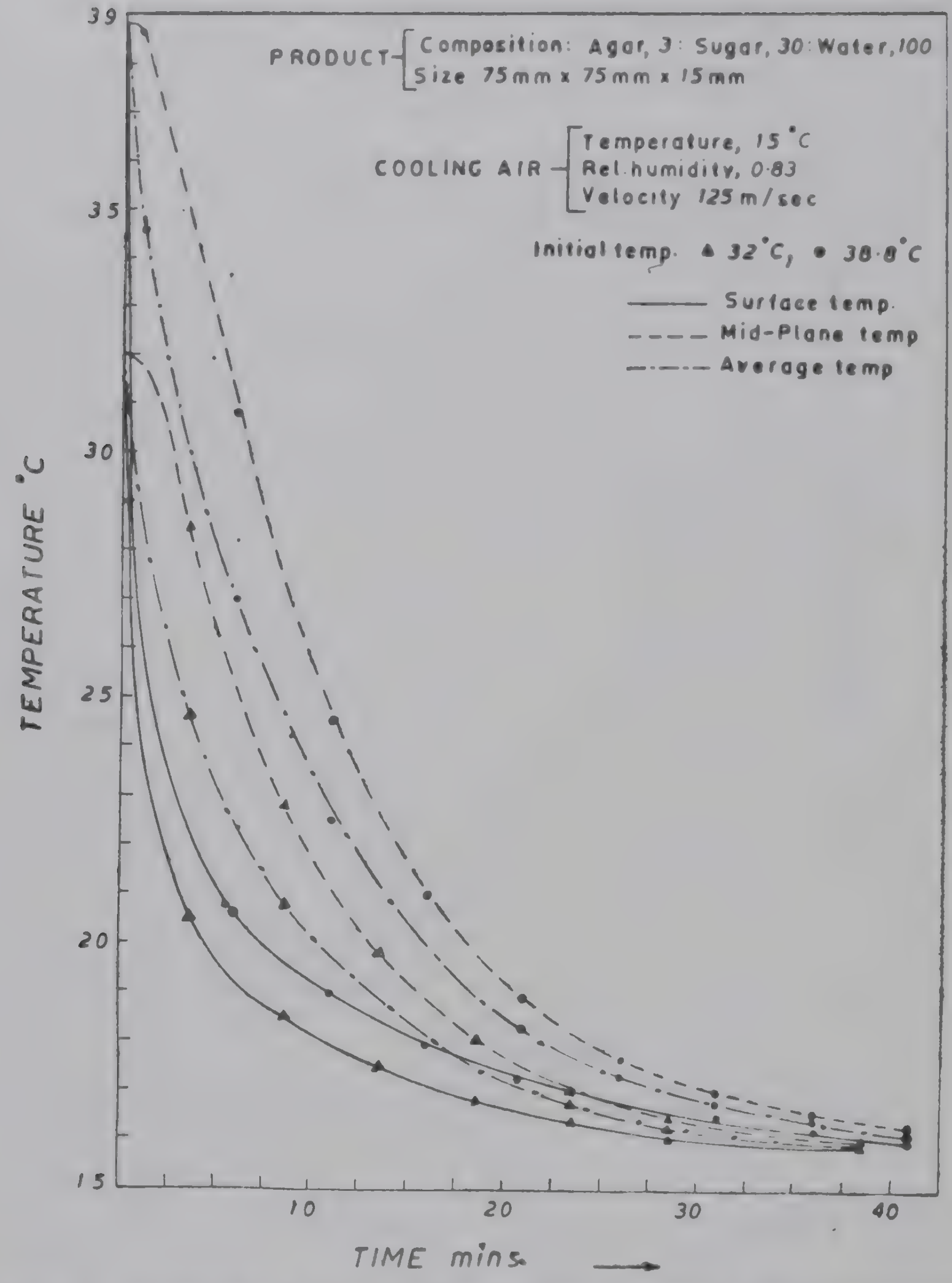

FIG. 5. Aircooling of model food gel — effect of product initial temperature

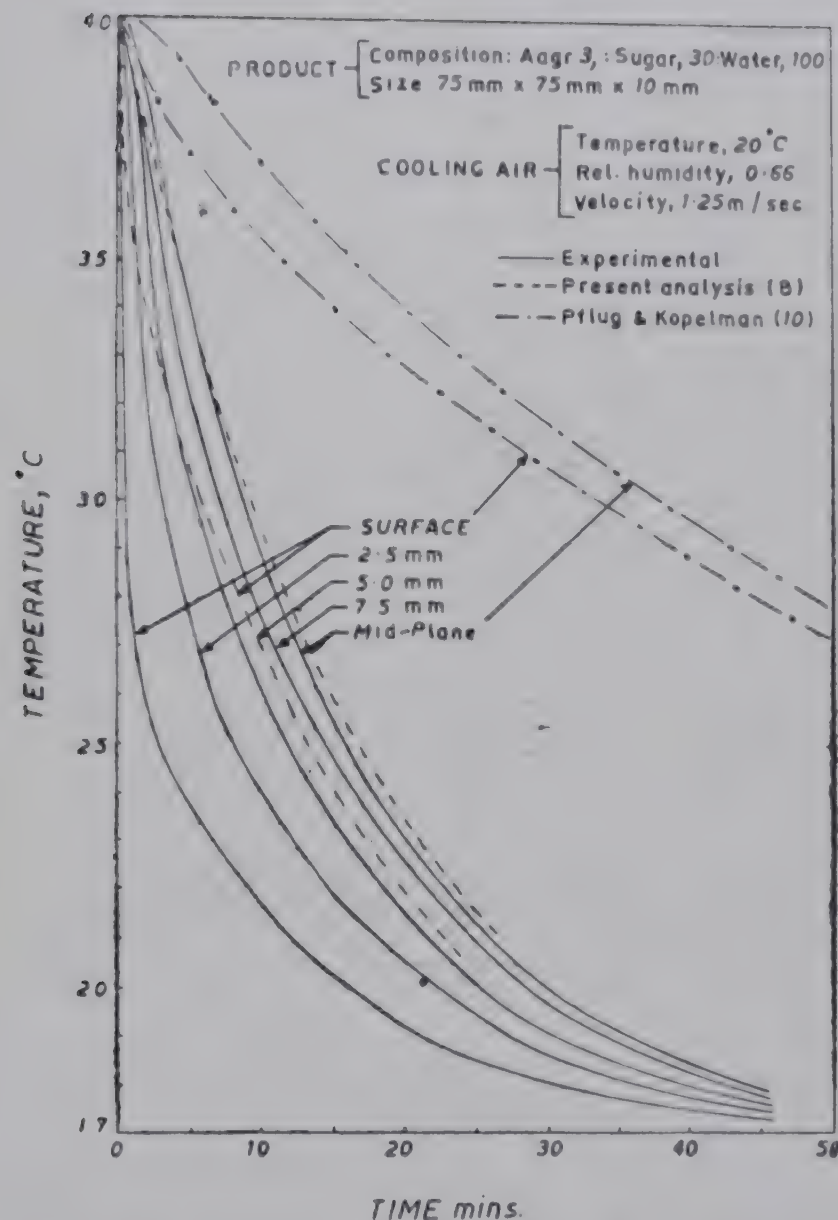

FIG. 6. Aircooling of model food gel — comparison with theoretical prediction

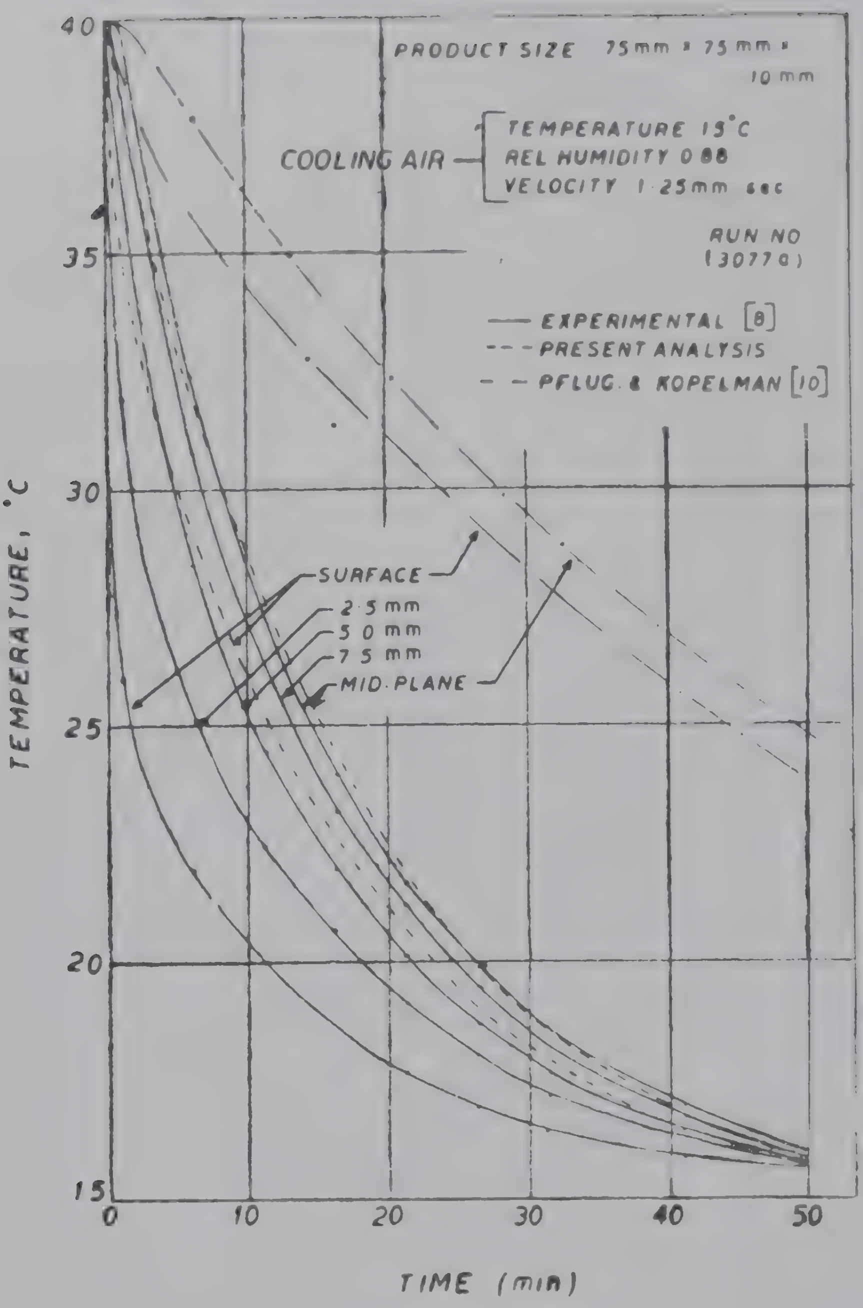

FIG. 7. Precooling of model food gel

predicted in this case, this trend being more significant at lower cooling air humidities. Also, the total processing time is overpredicted. Hence, the present practice of using conventional heat transfer charts for predicting the time-temperature characteristics and cooling loads yields erroneous results.

Some typical moisture loss curves are presented in Fig. 8. Sharp increase in moisture loss is observed at lower humidities. Higher velocities are also accompanied by higher moisture loss. Higher initial product temperature results in higher moisture loss in the earlier stages of processing. The moisture loss is minimum at low air temperatures. Fig. 9 shows the moisture loss curves obtained for

various compositions of the product. This result proves that the moisture loss is not purely a surface phenomena, but it also depends on the internal migration of moisture, thus depending on the product properties. In all the cases mentioned above, the

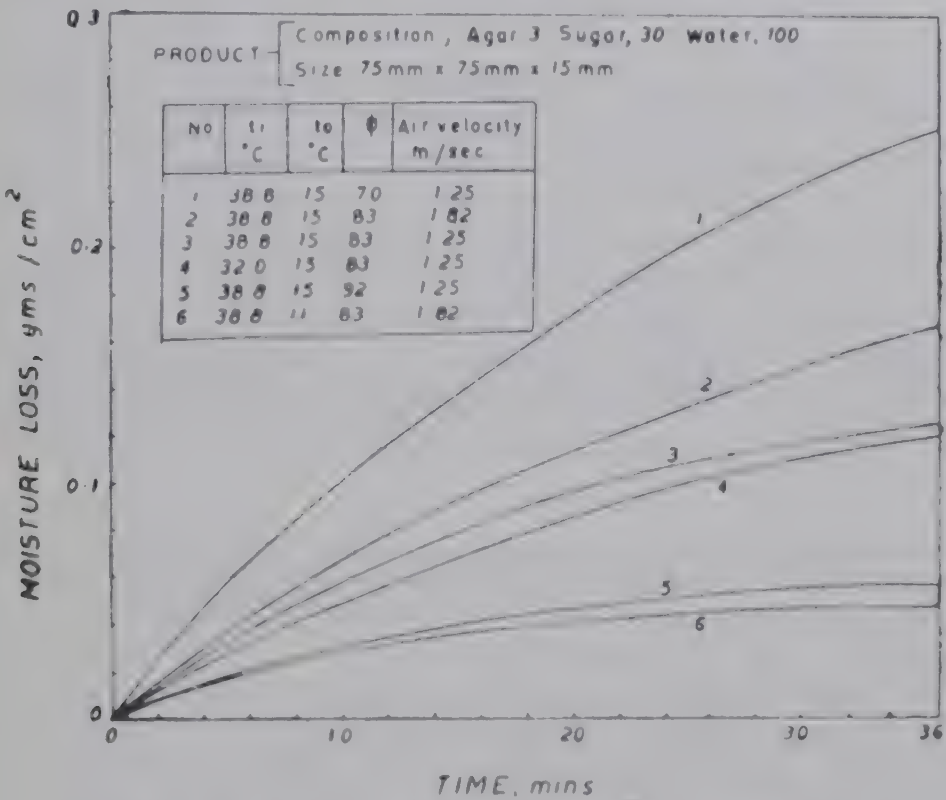

FIG. 8. Moisture loss during aircooling of model foods

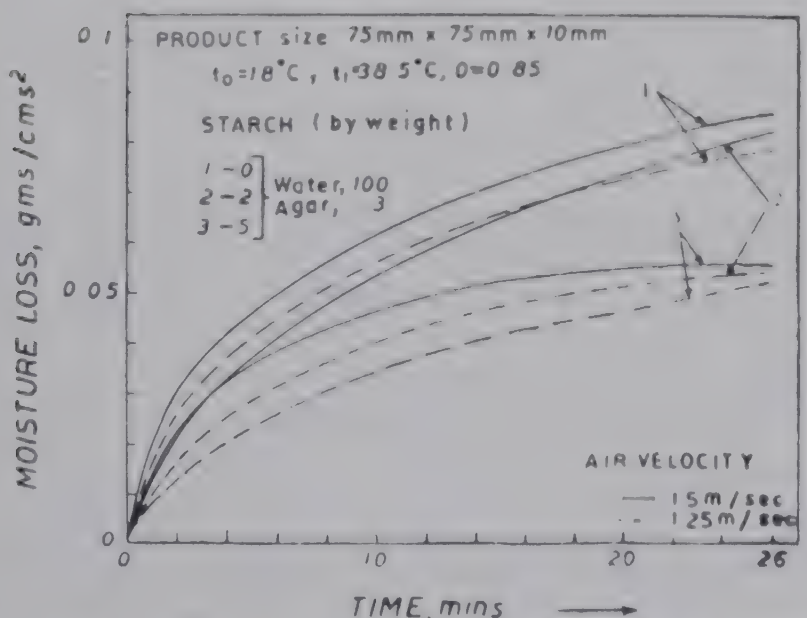

FIG. 9. Moisture loss during aircooling of model food gels effect of product composition

initial moisture transfer rate is high, which progressively decreases as the process continues. This is because the temperature differential and hence the partial water vapour pressure differential between the product surface and the ambient is large at the initial stages of the processing.

Summary

Experimental investigations on heat and mass transfer during aircooling of model foods are presented. The food models are constructed out of various proportions of soluble starch, sugar, Agar-Agar and water. They are cast in the form of slabs with embedded thermocouples and are cooled in an airconditioning tunnel. The parameters varied are the product composition and initial temperature and velocity, temperature and humidity of the cooling air. Moisture loss and time-temperature histories are measured. Cooling rates increase with increase in air velocity and lowering the humidity. However, these are accompanied by increased moisture loss. It is observed that the evaporation of moisture has a significant influence on heat transfer characteristics.

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Vapour-Liquid Equilibrium of Fluorocarbon Mixtures

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Introduction:

The use of mixtures in vapour-compression machines has many good points in its favour. The studies show the possibility of immediate application of mixed refrigerants to process refrigeration effecting power saving and also for low temperature refrigeration.

So far only mixtures of R12 and R22 have been studied[6,12,15,16,19,21]. A limited amount of experimental data, i.e. vapour-pressure data or equilibrium data have appeared in the literature[6–8,10,11,18,19,21].

The present paper describes the experimental set-up to obtain the equilibrium data. To start with, the mixtures of R12 and R13 are chosen with the view that these mixtures will have applications for low temperature refrigeration. The equilibrium data of six mixtures of R12 and R13 varying in composition from 5 per cent to 30 per cent of R13 in R12 in steps of about 5 per cent are measured. The vapour pressure of these mixtures are plotted on l_n P–1/T diagram and also the equation of the following form is fitted to the data of each mixture by the method of lease square using a computer programme,

$$l_n P = C_1 + \frac{C_2}{T} + C_3 l_n T + C_4 T$$

The computer programmes for the calculation of bubble-point temperature and dew-point temperature are developed and using these programmes the bubble-point temperature and dew-point temperature are calculated from the experimental data of R12–R13 mixtures and presented in the form of t—ξ and p—ξ diagrams.

Experimental setup

The measurement of equilibrium data or vapour-pressure data is essential to establish the vapour-liquid equilibrium and thereby arrive at the thermodynamic properties of mixtures.

The schematic arrangement of experimental setup used (Fig. 1) consists of (i) Vapour-liquid equilibrium still, (ii) Controlled temperature bath, (iii) Pressure measuring arrangement and (iv) Gas chromatograph with single channel recorder to analyse vapour phase compositions.

Equilibrium Still : Wichterle and Hala[22] have developed a rapid semi-micro method based on the sampling of very small volumes of the vapour phase which is then analysed by gas chromatograph. The present work makes use of a semi-micro static still for measurement of equilibrium data. The still is made of stainless steel of about 650 ml. capacity. It is tested upto 30 atmospheres of pressure. In the centre of the top cover a thermocouple pocket is screwed. A copper-constantan thermocouple is welded at the end of pocket which is always within the fluid. There are two more tappings (8 and 9) both of about 6 mm diameter welded to the calorimeter. Tapping '8' which is inserted only up to about 1/16th of the still height from the top so that it is always in the vapour phase, serves as pressure tapping as well as vapour sampling to gas chromatograph. Tapping 9 is used for charging the mixture.

Constant Temperature Bath : To maintain the constant temperature of the equilibrium still, a petrol bath is used. The bath is insulated by about 8 cm layer of glass wool. The bath temperature is regulated with the help of mercury-toluene regulator. A stirrer keeps the bath temperature uniform. With this arrangement bath temperature and thus the temperature of the mixture in the equilibrium still can be maintained constant with ±0·1°C accuracy for any length of time.

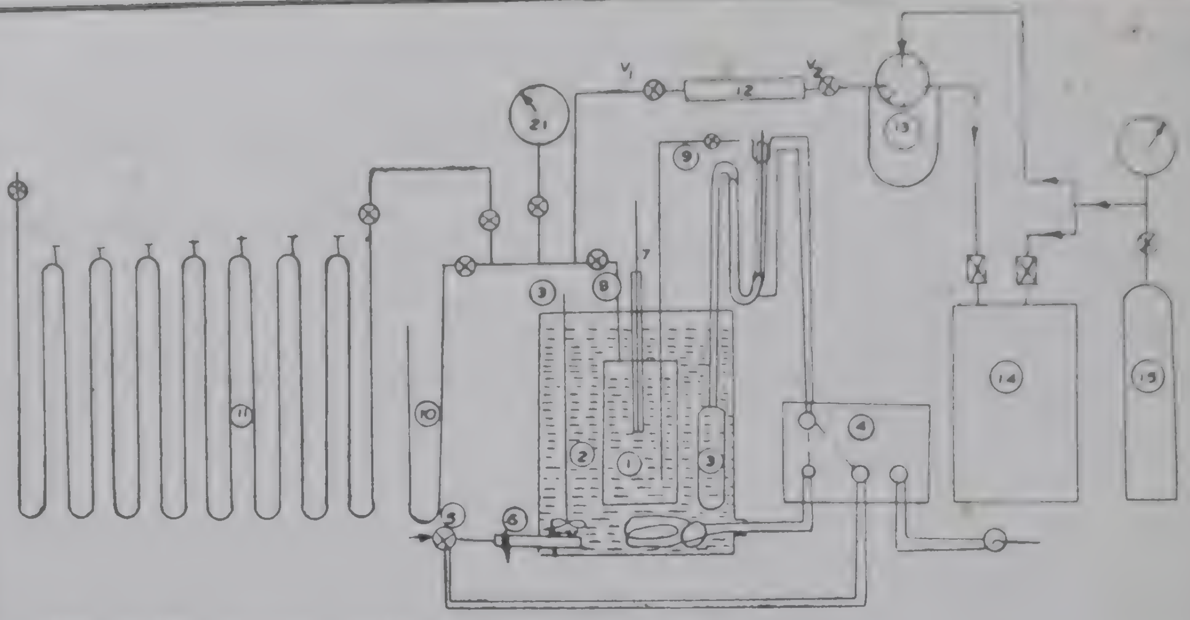

FIG. 1 BLOCK SCHEMATIC DIAGRAM OF EXPERMENTAL SET—UP.

(1) EQUILIBRIUM STILL; (2) CONSTANT TEMPERATURE BATH; (3) TEMPERATURE REGULATOR; (4) ELECTRIC RELAY; (5) SALENOID VALVE; (6) LIQUID AIR SUPPLY; (7) ELECTRIC HEATER; (8) PRESSURE TAPPING AND SAMPLING LINE; (9) CHARGING LINE; (10) SMALL RANGE MANOMETER; (11) SERIES COMPOUND MANOMETER; (12) GAS SAMPLE TRAP; (13) GAS SAMPLING VALVE WITH LOOP; (14) CHROMATOGRAPH UNIT (15) H_2 GAS CYLINDER.

The Temperature Regulator (Thermostat) : The temperature regulator is a controlling element which directly responds to the change in temperature. The temperature regulator developed for the purpose uses two fluids namely toluene and mercury. Toluene serves as temperature sensing fluid and mercury as the conducting fluid for making and breaking the electrical contacts.

The arrangement is shown in Fig. 2. The sensing bulb and connecting tubes are of pyrex glass. Toluene is filled up to point A to utilise the complete range of the temperature sensing device. The tube A to B is filled with mercury. The effective length of temperature sensing device is AC. The range of the thermostat depends on the length of the tube AC, and bulb size. The length of AC cannot be increased much because of handling difficulties. Two different thermostats are used to cover the whole range from —50°C to +50°C. The length of AC is kept 40 cm and the bulb size 56 ml as calculated on the basis of coefficient of volume expansion for toluene. This enables to regulate the bath temperature with an accuracy of ± 0·1°C.

An electrical actuator, i.e., relay and solenoid type control valve is used to bring in operation the alternate circuits for liquid air or the electric heater.

Pressure Measuring Arrangement : In the present work pressures are measured with the use of manometers. For low pressures a single U-tube manometer is used, while for high pressures a series compound mercury-water manometer is used. Mercury acts as manometric fluid while water as pressure communicating fluid.

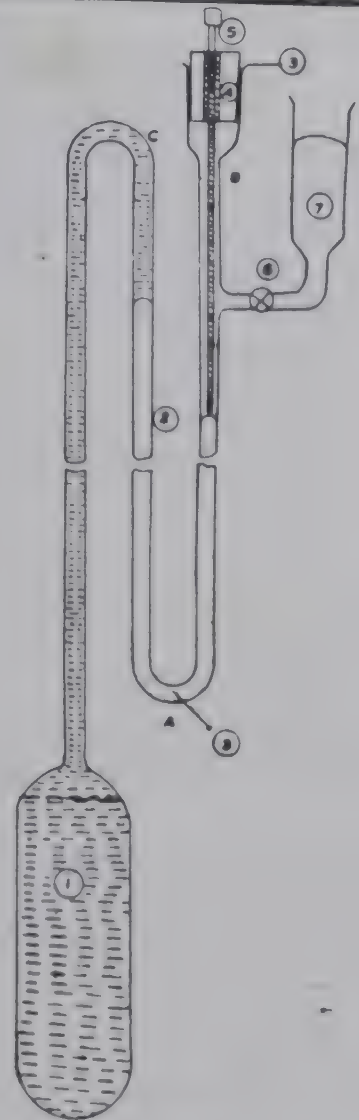

FIG. 2:- DIAGRAM OF TEMPERATURE REGULATER,
(1) SENSING BULB; (2) THERMOSTAT'S EFFECTIVE TUBE; (3) PLATINUM TERMINALS, (4) NUT; (5) THREADED SCREW; (6) STOP COCK, (7) MERCURY COMPENSATING RESERVOIR.

The manometer consists of eight U-tubes connected in series. Mercury is filled up to about half of each tube and the rest is filled with water. The first tube of the manometer and connecting line is perfectly evacuated before connecting to the pressure tapping of the equilibrium cell. To the readings of the manometer, the correction for weight of water column over the mercury and barometric pressure are applied. The use of this series-compound manometer achieves a high degree of accuracy in high pressure measurements.

Temperature Measuring Arrangement : The temperatures are measured with the use of copper-constantan thermocouples using a thermocouple potentiometer with an accuracy of 0.005 mV.

Gas Chromatograph Unit : The vapour phase compositions of the mixtures

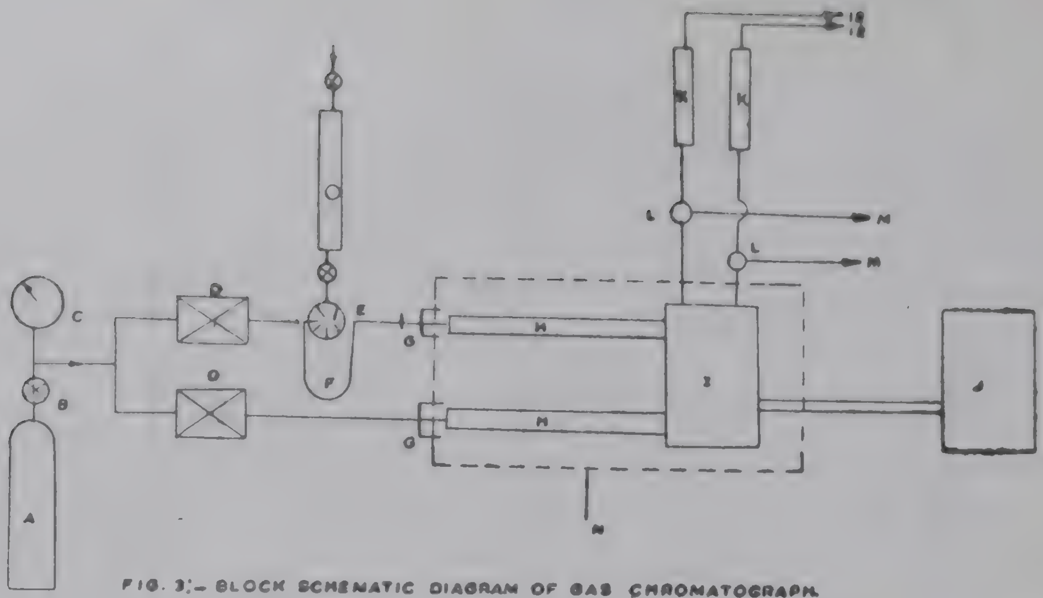

FIG. 3 :— BLOCK SCHEMATIC DIAGRAM OF GAS CHROMATOGRAPH.

(A) H_2 GAS CYLINDER; (B) PRESSURE REGULATING VALVE, (C) PRESSURE GAUGE; (D) NEEDLE VALVES; (E) GAS SAMPLING VALVE; (F) GAS SAMPLING LOOP, (G) INJECTION SITE; (H) SEPARATING COLUMNS; (I) DETECTOR HEAD; (J) m.v. RECORDER; (K) FLOW METERS; (L) 3WAY COCK (M) GAS OUTLETS; (N) CONSTANT TEMPERATURE OVEN; (O) GAS SAMPLE CULLECTOR.

are analysed using a Pye-Unicam gas chromatograph equipped with heated katharometer type detector along with katharometer power supply Unit, Detector oven Controller, Programmer, Controller and a Cambridge recorder (lmV). This gas chromatograph is provided with sampling valve and different capacity gas sampling loops for proper sampling (Fig. 3).

Measurement Technique and Experimental Data

The equilibrium still is first evacuated and is filled with the required mixture. The total mixture charged in the cylinder is about 700 gms so that about 7/8th of the cell volume is occupied by the liquid phase of the mixture. Consequently, the volume occupied by vapour phase is very small. Thus the liquid's composition can be approximated to overall composition and also the change of volume on mixing can be neglected[20] without introducing sufficient error.

The temperature of bath and also of the mixture in equilibrium still is first lowered to about —70°C by adding liquid air to the petrol directly. Then the required temperature is maintained in the bath with the use of temperature regulator, by adjusting the threaded screw.

Once the required temperature is achieved and remains constant for about 15 to 20 minutes which is ensured by measuring the temperature and vapour pressure of the mixture, the measurements of temperature and pressure are made.

Thus the equilibrium pressure and temperature are known. Finally the vapour which is in equilibrium with the liquid at the measured temperature and pressure is trapped between the two valves V_1 and V_2 (Fig. 1) to analyse the vapour phase

composition. The sampling valve, sampling loop and other connecting lines are evacuated and then the sampling loop knob is put on FILL and the valve V_2 is open; thus a measured volume of the sample is filled in the sampling loop. This measured quantity of the sample is carried away by the carrier gas to the gas chromatograph column by turning the sampling valve knob to INJECT. The gas chromatograph analyses the sample and gives an electrical signal to *mV* recorder proportioned to the composition of the two components. This way the temperature, pressure and composition of vapour phase are measured. Again the threaded screw of the temperature regulator is adjusted to another temperature and the above measurements are repeated.

To check the reliability and accuracy of the measurements the vapour-pressure data of pure R12 are measured twice during the experiment. These data compared

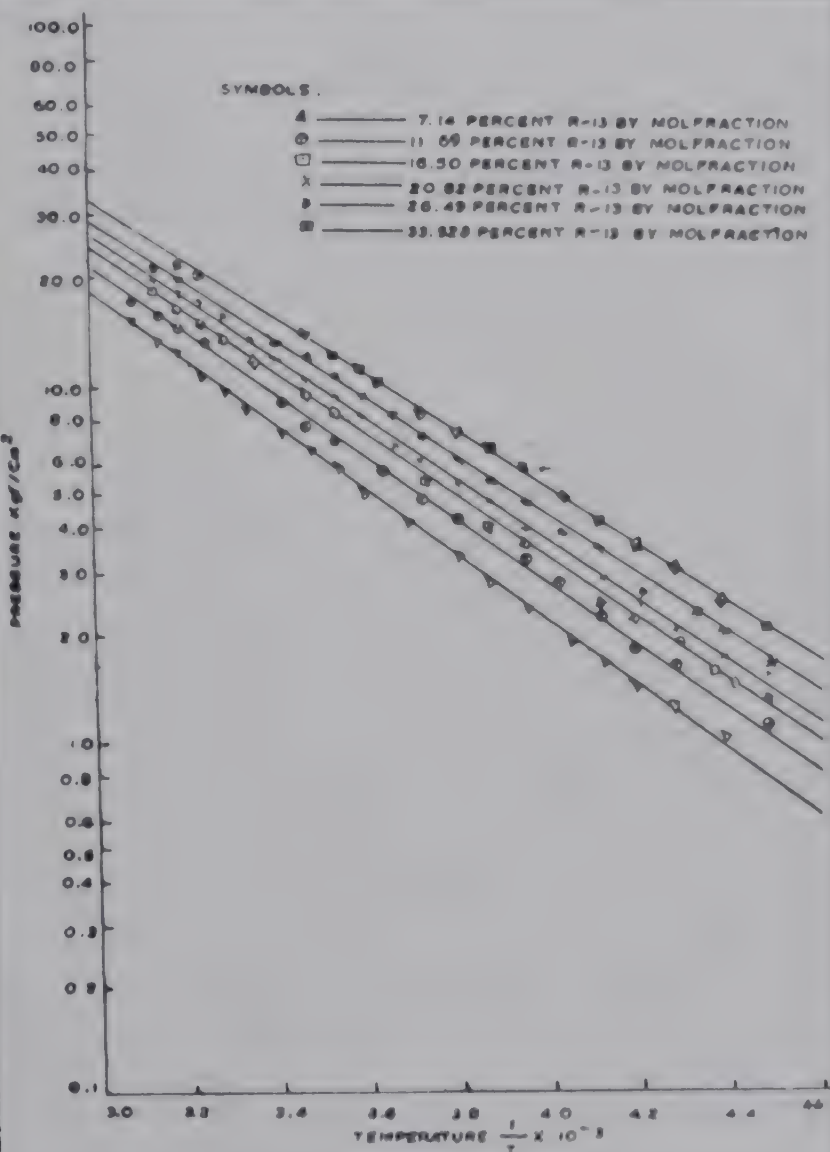

FIG. 4 L_n P - $\frac{1}{T}$ DIAGRAM FOR THE MIXTURES OF R-12 & R-13

within an accuracy of 1.4 per cent with the standard tables. Further to calibrate and to check the reliability of the gas chromatograph the mixtures of vapour-phase of known composition were analysed by taking a number of observations which show consistency in the measurement within 0.5 per cent.

The equilibrium data are measured for a wide range of temperature from —50°C to + 50°C at an interval of about 5°C for each mixture.

The vapour-pressure obtained are presented in the form of $l_n P - \frac{1}{T}$ diagram in Fig. 4. Further the equation of the following form is fitted to these data by computer programme.

$$l_n P = C_1 + \frac{C_2}{T} + C_3 l_n T + C_4 T$$

The constants C_1, C_2, C_3 and C_4 are tabulated in Table 1 for the six mixtures.

TABLE 1

Mole percentage of R13	C_1	C_2	C_3	C_4
7.14	−147.73382	2158.9131	27.250233	-4.2289632×10^{-2}
11.59	481.61561	−13172.292	−84.706123	1.5942455×10^{-1}
16.50	−565.10494	12260.596	101.81767	-1.7990262×10^{-1}
20.82	−591.21341	12685.692	106.85183	-1.929236×10^{-1}
26.49	−488.54203	9793.4396	89.295664	-1.6892074×10^{-1}
33.53	30.546517	−2577.5847	−3.4584887	$2\ 1195040\times 10^{-3}$

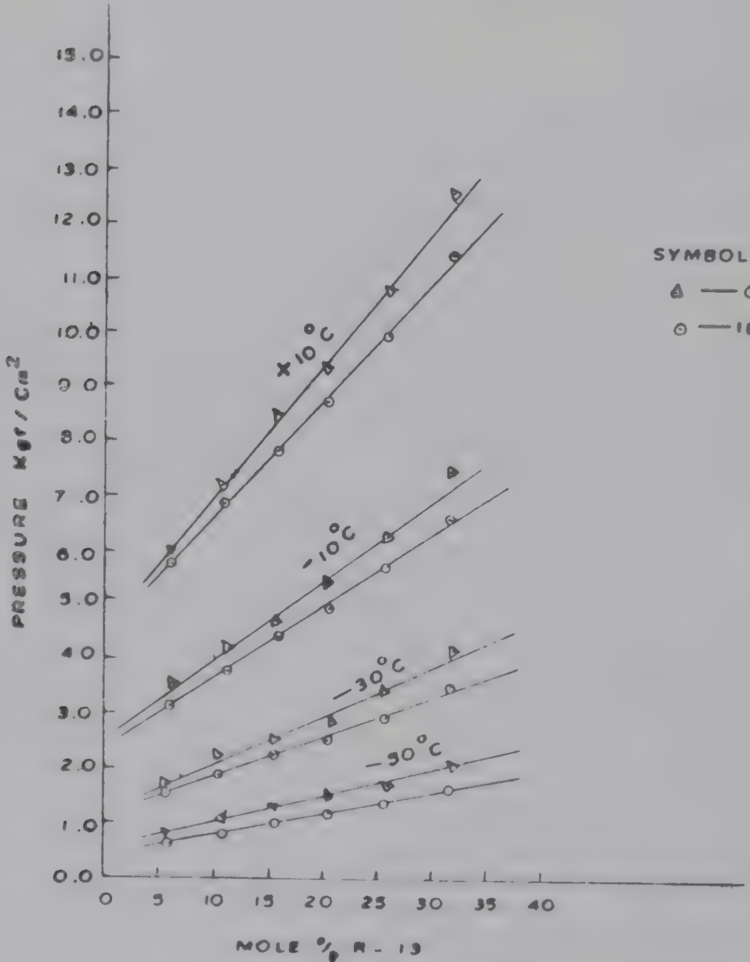

FIG. 5

Results and Discussions

The vapour pressure data measured and calculated by Raoult's law.

$$P_I = X_{12}P_{12}^{\circ} + X_{13}P_{13}^{\circ}$$

where X_{12} and X_{13} are mole fractions of R12 and R13 respectively and P°_{12} and P°_{13} the saturation pressures of pure components, are plotted as a function of composition in Fig. 5. Also the deviations from these ideal pressures are calculated as follows :

$$\% \text{ Deviation} = \frac{(P_{obs} - P_I)}{P_I} \times 100$$

The resulting deviations at different temperatures are shown in Fig. 6.

All the mixtures of R12 and R13 show a positive deviation from Raoult's law. The percentage deviation decreases with increase in temperature from about 27 per cent at —50°C to about 4 per cent at +30°C. This is because, the intermolecular forces contribute to non ideality and become less influential at higher temperatures.

Further, the computer programmes, for calculating the bubble-point temperatures and dew-point temperatures are developed based on the theory discussed in an earlier paper[2]. The bubble-temperatures and dew-temperatures are calculated with the use of these programmes for the above mixtures of R12 and R13. The t — ξ and P — ξ diagrams are plotted as shown in Figs. 7 and 8.

These diagrams are necessary in developing the other thermodynamic properties, viz., enthalpy and entropy of the mixture and hence to design and optimise the system.

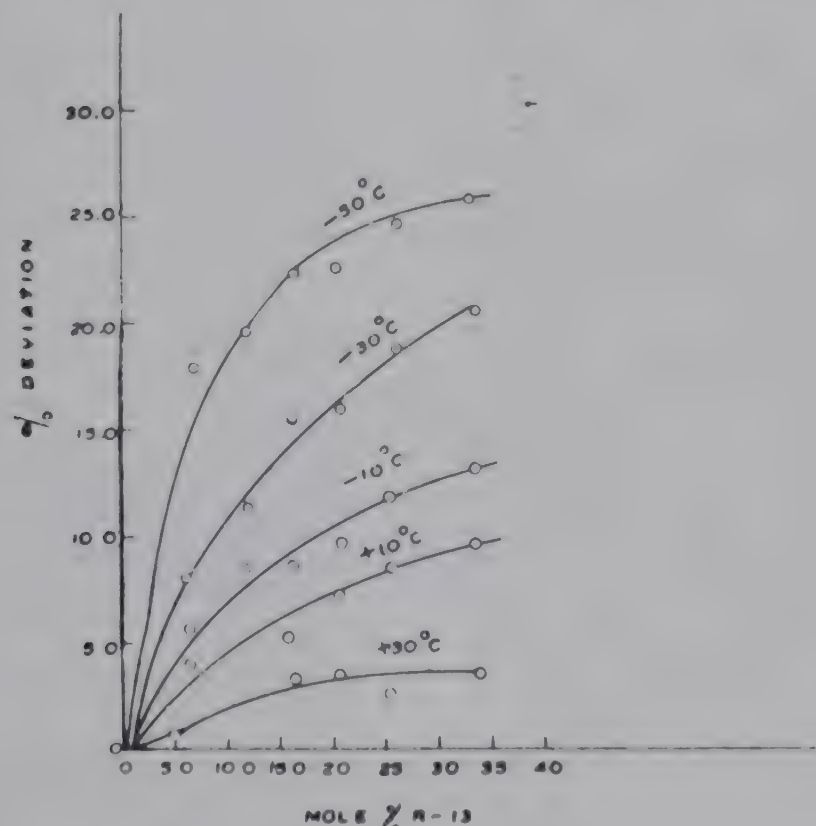

FIG. 6

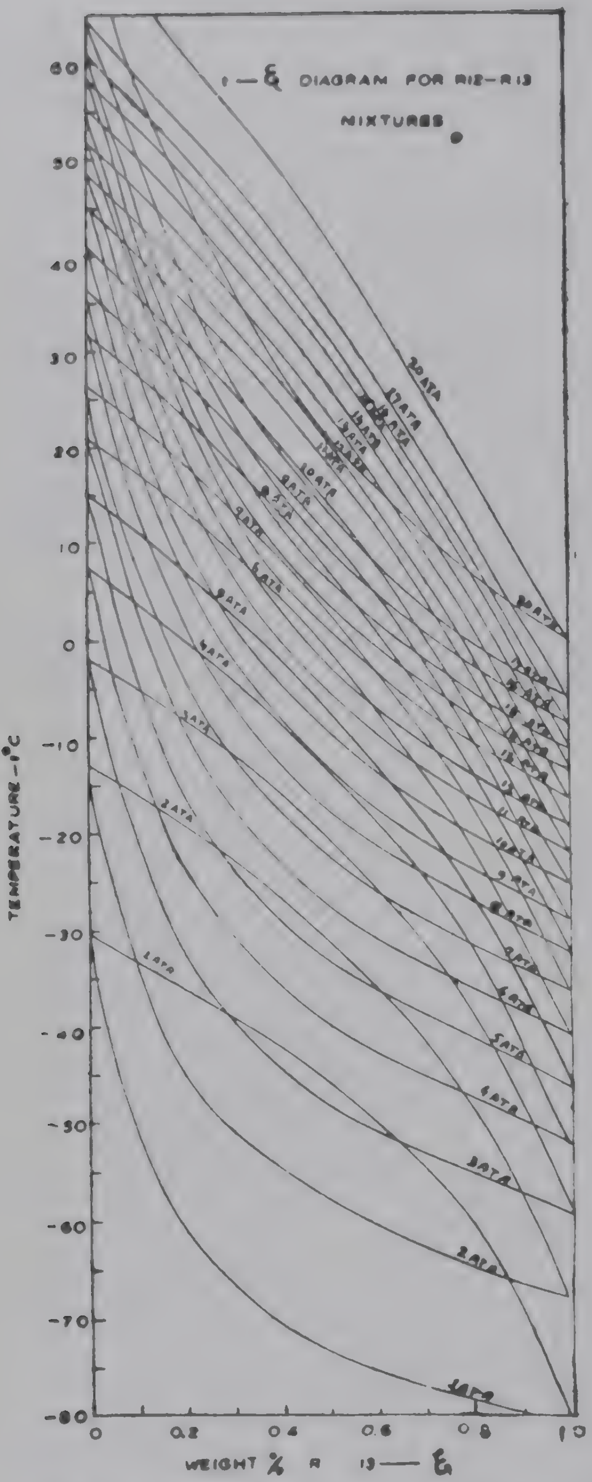

FIG. 7. t–ξ Diagram for R12–R13 Mixtures

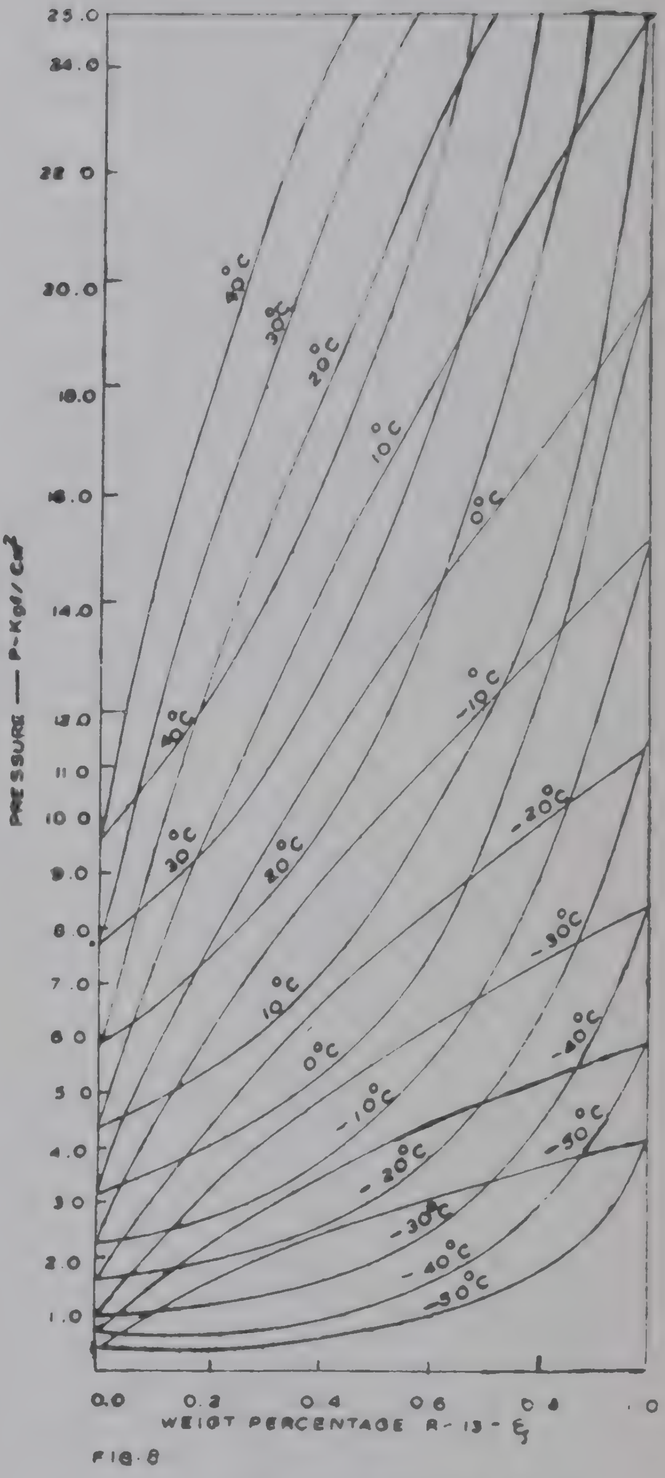

FIG. 8. t–ξ Diagram for R12–R13 Mixtures

Considerable interest has developed in recent years in the use of mixed refrigerants in vapour-compression machines. However, not much work has been done on the thermodynamic properties of mixtures of fluorocarbon refrigerants.

Summary

The present paper is a part of the attempt to develop an experimental and computer method to establish these properties for mixtures of fluorocarbons in general. It discusses the experimental set up used for obtaining the equilibrium data of mixtures of R12 and R13 and presents the various equilibrium properties of these mixtures calculated from their equilibrium data.

The experiment consisted of the measurement of equilibrium data of six mixtures of R12 and R13 varying in composition.

Computer programmes have been developed to calculate the bubble-point and dew-point temperatures. A number of important plots, viz., $l_n P - \frac{1}{T}$, $t - \xi$ and $P - \xi$ diagrams are obtained for the above mixtures.

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Estimation of Thermodynamic Properties of R–22 and Dimethyl Formamide Mixtures

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The rapid development and diversification of the refrigeration and air conditioning industries in recent years has led to an increasing interest in refrigerants specially suitable for particular applications. The common halogenated hydrocarbon refrigerants are limited in number. Special characteristics of desired refrigerants, not possessed by individual refrigerants, lead us to the study of mixtures.

One difficulty with regard to the use of mixtures is that thermodynamic data are seldom available. However, if the mixture obeys Raoult's law reasonably well, properties of the mixture can be estimated from the data of its components. But, mixtures of nonpolar chlorofluorocarbon refrigerants with each other (like R 12 with R 114) or of polar chlorofluorohydrocarbon refrigerants with each other (like R 22 with R 21) obey Raoult's law approximately.

When a polar and a nonpolar refrigerants are mixed, the deviation from the Raoult's law is large. A mixture is considered suitable for vapour compression or for vapour absorption refrigeration system, if it gives large positive or large negative deviation from Raoult's law.

Eiseman[1] has very clearly illustrated the requirements for an ideal absorption refrigerant, such as extremely high chemical stability, low pressure, low ferric decomposition, long life and good thermodynamic properties.

Zellhoefer and others[2,3] measured the solubilities of several halogenated methanes in many organic solvents. According to their findings, halogenated methanes which contain a hydrogen atom in the molecule produced excess solubility in organic solvents in many cases. This excess solubility was attributed to "hydrogen bonding", which is essentially an electrostatic attraction between molecules.[4] Other investigators[1,5,6,7] have used this principle to seek a good solvent-refrigerant combination for the absorption cooling cycle.

The most promising solvents studied by Albright[7–10] were the di-N substituted formamides. Both diethyl formamide (DEF) and dimethyl formamide (DMF) showed high negative deviation from Raoult's law with either R 21 or R 22. On a mole basis DEF dissolved more refrigerant than DMF, but on weight basis DMF was a slightly better solvent. It should be noted that DMF—R 11 and DEF—R 21 showed positive deviation from Raoult's law. A comparison was also made of the solvent capacities of DMF and DEF with those of dimethyl ether of tetraethylene

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glycol (DME—TEG), which has been suggested as a good solvent for the halogenated methanes[6]. On a mole basis DME—TEG is best, but on weight basis, DMF and DEF are better solvents than DME—TEG.

Considering these advantages the thermodynamic properties of R 22 and DMF mixtures have been estimated using Albright's P—T—X data[7].

Critical Properties for the Components :

The following critical properties for R—22 and DMF were available in the literature[11,12]

	R—22	*DMF*
Critical temperature	= 369.33°K	647°K
Critical pressure	= 49.247 atm	44.2 atm
Critical volume	= 1.9493 dm³/kg	265cc/gm mol
Critical compressibility	= 0.274	0.218

Properties of the Pure Components :

The vapour pressure data for R—22, reported by Zander[11], was fitted through a computer programme, and the following relation was established

$$\text{Log}_e \text{ Pata} = A + B/T_1 + C/T_1^2 + D/T_1^3 + E/T_1^4 + F/T_1^5$$

where T_1 = T in °K/1000.0 (1)

A = 21.494002
B = —16.453023
C = 7.006649
D = —1.7224115
E = 0.20987229
F = —0.010254912

For DMF, Delzenne[13] and Gallant[14] have reported the vapour pressure data, the values are in good agreement with each other. Through a computer programme, Gallant's data, was fitted into the following relation

$$\text{Log}_e \text{ Pata} = A + B/T_1 + C/T_1^2 + D/T_1^3 + E/T_1^4 + F/T_1^5$$

where T_1 = T in °K/1000.00 (2)

A = 429.006860
B = —484.147980
C = 121.1814300
D = 46.5180450
E = —26.1698480
F = 3.2640684

The following data for *R—22* were used :

Molecular weight = 86.469
Normal boiling point = 233°K
Liquid Specific Heat is given by the following relation

$$Cp(\text{Cal/gm°K}) = 0.267165 + 3.22637 \times 10^{-5}\,T - 1.93112 \times 10^{-6}\,T^2 + 7.26552 \times 10^{-9}\,T^3 \quad (3)$$

Ideal Gas Specific Heat between 100°K and 590°K,

$$Cp^{\circ}(\text{Cal/gm}^{\circ}\text{K}) = 0.0712690 + 2.41204 \times 10^{-4}\,T - 2.4287 \times 10^{-7}\,T^{2} - 3.99893 \times 10^{-10}\,T \quad (4)$$

For *dimethyl formamide,* Gallant's data were fitted, through a computer programme and following relations were obtained

Heat of Vapourization

$$H_v = a + bt + ct^2 + dt^3 \quad (5)$$

where t is in °C and H_v in Cals/gm.

a = 164.180397
b = —0.282849
c = 0.001315
d = —0.000004

Vapour Heat Capacity

$$Cp^{\circ}_v = a + bt + Ct^2 \quad (6)$$

where t is in °C and Cp°_v in Cals/gm °C

a = 0.290000
b = 0.000795
c = —0.0000003

Liquid Head Capacity

$$Cp_L = a + bt + ct^2 \quad (7)$$

where t is in °C and Cp_L in Cals/gm °C

a = 0.491414
b = 0.000076
c = 0.000004

Equation of State : The Redlich-Kwong equation[16] of state, which was used for the estimation of properties, is given as

$$P = \frac{RT}{(v - b)} - \frac{a}{T^{1/2}\,V.(v + b)} \quad (8)$$

where $a = 0.4278\,R^2\,T_c^{2.5}/P_c$ (8a)

$b = 0.0867\,R\,T_c/P_c$ (8b)

The above equation can also be expressed as

$$Z^3 - Z^2 + (A^2P - BP - B^2P^2)\,Z - A^2\,BP^2 = 0 \quad (9)$$

where $A^2 = a/R^2\,T^{2.5}$ (9a)

$B = b/RT$ (9b)

$Z = PV/RT$ (9c)

Equation (9) was solved, using a computer programme and values of compressibility factor were obtained at the system temperature and pressure, for the two pure components.

Vapour-liquid Equilibrium

Ljunglin and Van Ness[17] have presented thermodynamically exact general equations applicable either at constant temperature or constant pressure data for binary systems. This method allows the direct calculation of vapour composition from P—T—X data. The relation connecting the variables is called the general coexistence equation, because it is an equation which must be satisfied when phases coexist at equilibrium.

The derivation of this equation is based on the following equation, given by Van Ness[18], (which is valid for liquid and vapour phase whether in equilibrium with another phase or not) :

$$\frac{V}{RT}dp + \frac{(H - H)}{RT^2}dT = X_1\, dln\,\overline{f_1} - X_2\, dln\,\overline{f_2} \tag{10}$$

If equation (10) is applied first for vapour phase and then for liquid and the resulting relations are combined by subtraction, because the conditions for equilibrium require that the pressure, temperature and fugacities should be identical, we get the following equation (11), which is a completely rigorous general coexistence equation :

$$\psi dp + \Omega\, dT = (Y_1 - X_1)\, d\, ln\frac{Y_1{}^V}{Y_2{}^V} + \frac{Y_1 - X_1}{Y_1(1 - Y_1)}dY_1 \tag{11}$$

$$\text{where } \psi = \frac{\triangle V^V + X_1 V_1{}^V + X_2 V_2{}^V - V^L}{RT} \tag{11a}$$

$$\Omega = \frac{-(\triangle H^V + X_1 H_1{}^V + X_2 H_2{}^V - H^L)}{RT^2} \tag{11b}$$

This equation can be integrated either for constant temperature or for constant pressure case. As, in this study, $P—X$ data at constant temperature of Thieme and Albright (which is given in Table 1) was used, the equation (11) can be simplified to[17] :

$$\frac{dY_1}{dP} = \frac{\psi - (1 - 2Y_1)(Y_1 - X_1)(\delta/RT)}{[(Y_1 - X_1)/Y_1(1 - Y_1)] - (Y_1 - X_1)(2P\delta/RT)} \tag{12}$$

$$\text{where } \delta = 2B_{12} - B_{11} - B_{22} \tag{12a}$$

Second virial coefficients for the two components B_{11} and B_{12} were calculated using Redlich-Kwong equation of state as follows[19]

$$B(T) = b - a/RT^{3/2} \tag{13}$$

where a and b are $R—K$ constants, as stated earlier.

For value of δ, one must have the interaction coefficient B_{12}. But, due to lack of data for *DMF*, it was not possible to calculate B_{12}, using Stockmeyer's potential functions. Under these limitations, ideal vapour phase conditions were assumed, as discussed by Prausnitz and Gunn[20] i.e.

$$B_{12} = (B\,11 + B\,22)/2 \tag{14}$$

Thus, equation (12) was numerically integrated, to obtain vapour composition, using Runge-Kutta method, through a computer programme, and the computed values were plotted in the form of pressure-composition and temperature-composition curves for R-22 and *DMF* mixtures.

Table I. P—T—X data for R—22/DMF Mixture

Temp °K	Mole Fraction R—22	Pressure ata
310.78	0.000	0.0102*
	0.212	1.3266
	0.483	3.8095
	0.639	6.5351
	1.000	14.5000*
324.67	0.000	0.0218*
	0.209	1.9047
	0.480	5.4082
	0.627	8.8435
	1.000	20.0000*
338.56	0.000	0.0470*
	0.205	2.6530
	0.475	7.5510
	0.617	11.6327
	1.000	26.800*
352.45	0.000	0.0850*
	0.201	3.5374
	0.470	9.9319
	0.595	14.6938
	1.000	35.5000*
366.33	0.000	0.1430*
	0.196	4.5918
	0.465	12.6870
	0.577	18.0612
	1.000	46.8000*
380.18	0.000	0.2458*
	0.189	5.8503
	0.458	15.9523
	1.000	60.4000*
394.10	0.000	0.4160*
	0.182	7.3129
	0.452	19.5918
	1.000	78.2000*

* Values not given by Thieme and Albright, but obtained from vapour pressure equation for the respective pure component.

Properties of the mixture :

For R–K equation of mixture the following combination Rule[21] was used.

$$A = \sum_{i=1}^{n} Y_i A_i \tag{15a}$$

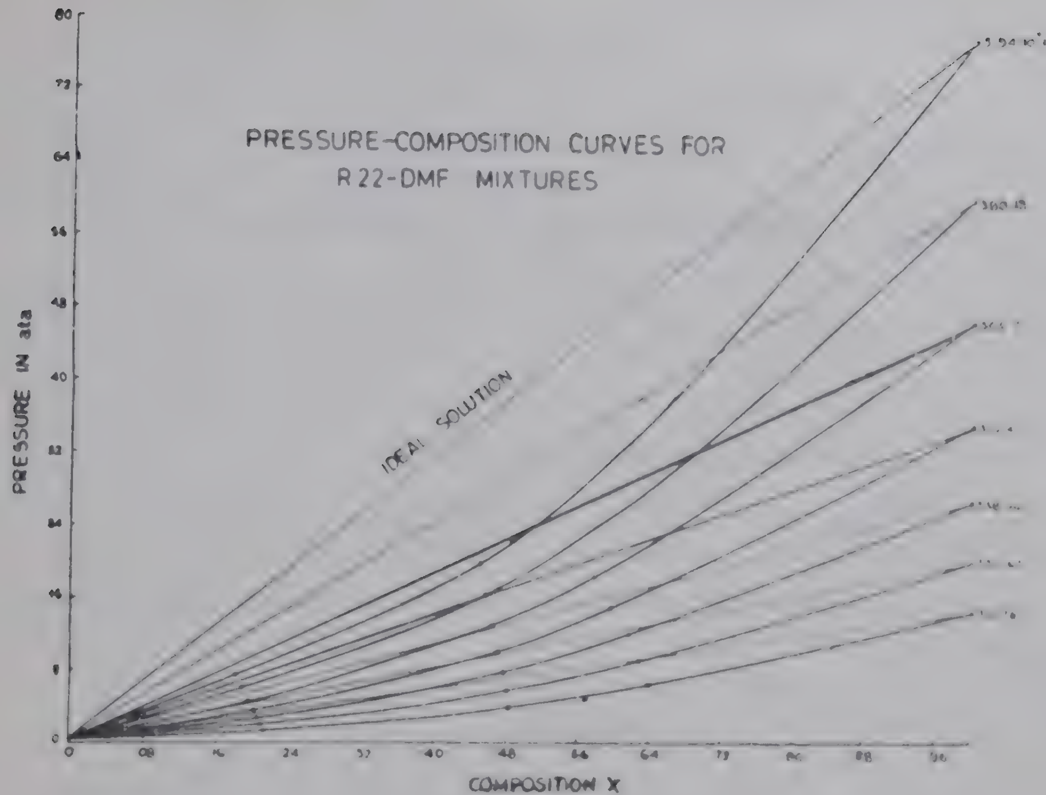
PRESSURE-COMPOSITION CURVES FOR
R 22-DMF MIXTURES
IDEAL SOLUTION
PRESSURE IN ata
COMPOSITION X

FIG. 1

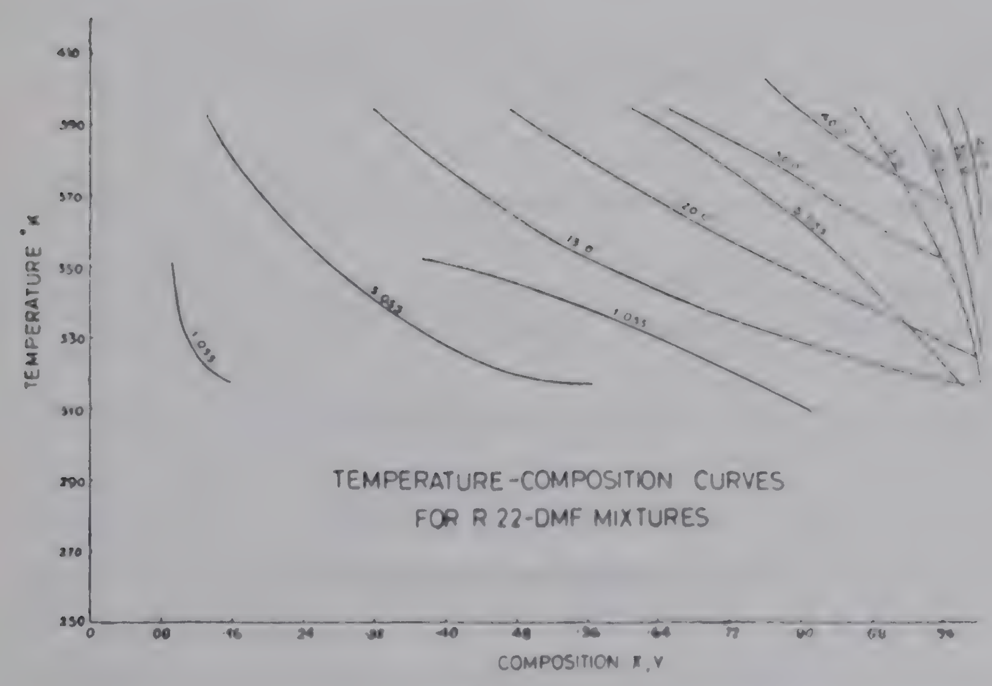
TEMPERATURE-COMPOSITION CURVES
FOR R 22-DMF MIXTURES
TEMPERATURE °K
COMPOSITION X,Y

FIG. 2

$$B = \sum_{i=1}^{n} Y_i B_i \tag{15b}$$

The Gas phase fugacity coefficients of R–K equation are given by the following relation

$$ln\phi_i = (Z-1)\frac{Bi}{B} - ln(Z-BP) - \frac{A^2}{B}\left(\frac{2Ai}{A} - \frac{Bi}{B}\right) ln\left(1 + \frac{BP}{Z}\right) \tag{16}$$

The effect of pressure on enthalpy is given by[16]

$$\left(\frac{d\, ln\, \phi i}{dT}\right)_{p,y} = \left(\frac{Hi^{O} - Hi^{-P}}{RT^2}\right) = -\frac{\Delta \bar{H}i}{RT^2} \tag{17}$$

Equation (16) can be differentiated to obtain an analytic expression for partial enthalpies.[21]

$$\frac{-\Delta \bar{H}_i}{RT^2} = \frac{B_i}{B}\left(\frac{dZ}{dT}\right)_{P,y} - \frac{1}{(Z-BP)}\left[\left(\frac{dZ}{dT}\right)_{P,y} - P\left(\frac{dB}{dT}\right)_{P,y}\right]$$
$$-\left[2\frac{Ai}{A} - \frac{Bi}{B}\right]\left[ln(1+BP/Z)\right]\left[\frac{d(A^2/B)}{dT}\right]_{P,Y} \frac{-A^2}{B}\left(\frac{2Ai}{A} - \frac{Bi}{B}\right)$$
$$\left[\frac{-\frac{P}{Z}\left(\frac{dB}{dT}\right)_{P,Y} \frac{-BP}{Z^2}\left(\frac{dZ}{dT}\right)_{P,y}}{1 + BP/Z}\right] \tag{18}$$

In equation (18) all the partial derivatives can be evaluated with the help of R–K equation, and thus partial enthalpies for the mixture were computed.

Enthalpy Calculations

The following procedure was adopted, and the subscripts refer to the P–H diagram shown :

(i) A base enthalpy value of $H_1 = 100$ K Cal/Kg for the saturated liquid at pressure corresponding to temperature 0°C was assumed for each pure component.

(ii) In the vapour state, point 3 i.e. at P = 0, t = 0°C is the reference point. Therefore,

$$H_3 = H_1 + L_{Pt=0} + \left[\begin{array}{l}\text{Isothermal Enthaphy Departure}\\ \text{term using } R\text{–}K \text{ Equation}\end{array}\right]_{P=0}^{P_{t=0}}$$

$$= H_1 + LP_{t=0} + \left[-RT\left\{\frac{3A^2}{2B}\, ln\left(1 + \frac{BP}{Z}\right) + (1-Z)\right\}\right] \tag{19}$$

Thus, the two reference points (liquid and vapour) were defined and inter-related.

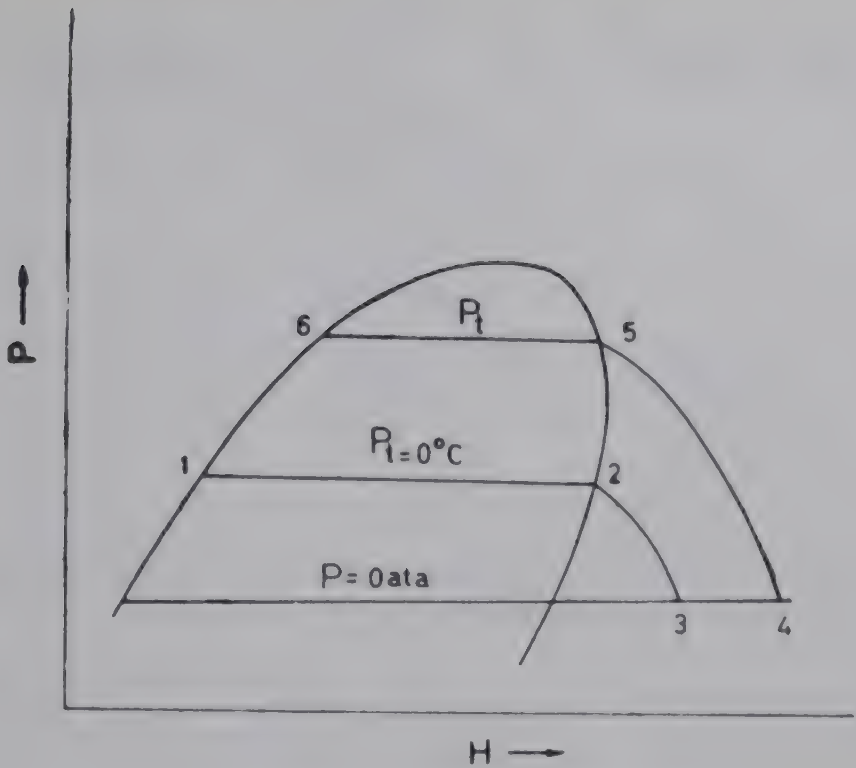

Fig. 3. P—H diagram

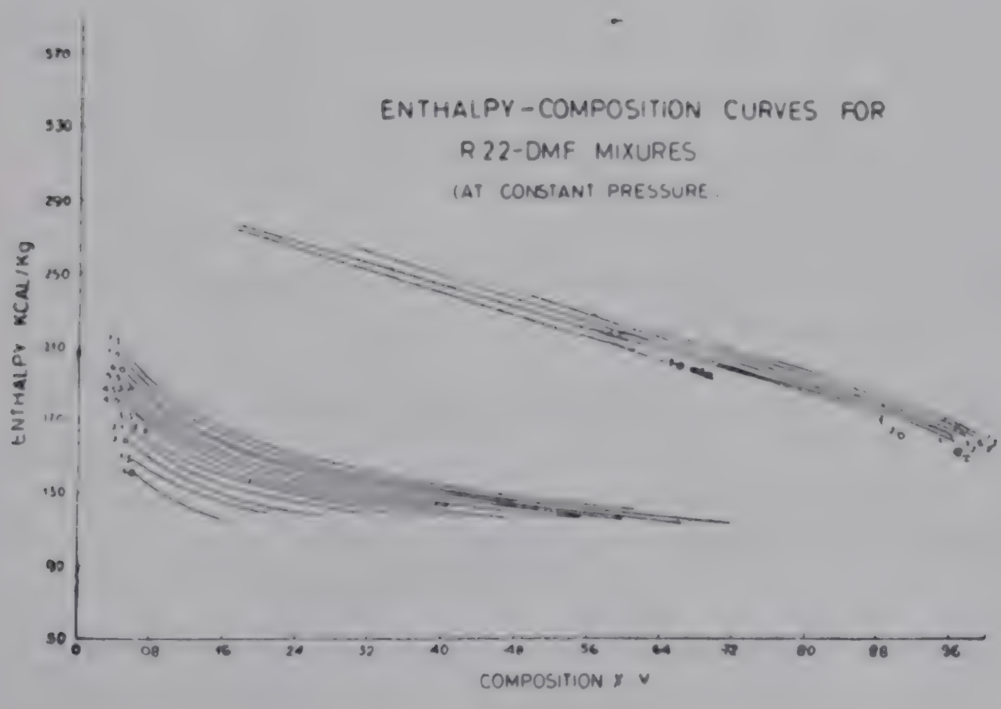

Fig. 4

(iii) This zero reference pressure being assumed, the enthalpies of the pure components are first calculated at the system temperature and reference pressure from

$$H\,T_i^{\circ} - H^{\circ}_{Tref_i} = \int_{Tref}^{T} Cp_i^{\circ}\, dT \tag{20}$$

where Cp_i° are already known for the respective pure components.

(iv) At the temperature T, $P = 0$, the mixture enthalpy H_m° is determined per mole from equation

$$H_m^{\circ} = \sum Y_i \,.\, H_{Ti}^{\circ} \tag{21}$$

(v) The mixture enthalpy at the system pressure is then determined by calculating the isothermal enthalpy deviation

$$H_m^{v} = (H_m - H_m^{\circ}) + H_m^{\circ} \tag{22}$$

where the enthalpy deviation $(H_m - H_m^{\circ})$ was determined from equation (18), as mentioned earlier. The final resulting equation for the saturated vapour enthalpy is

$$H_m^{v} = (H_m - H_m^{\circ}) + Y_1 \left[100.00 + (H_3 - H_1)_1 + \int_{Tref}^{T} Cp_1^{\circ}\, dT \right]$$

$$+ Y_2 \left[100.00 + (H_3 - H_1)_2 + \int_{Tref}^{T} Cp_2^{\circ}\, dT \right] \tag{23}$$

(vi) The saturated liquid enthalpy at the system temperature and pressure is given by

$$H_m^{L} = \sum_i X_i H_i^{L} - \sum X_i RT^2 \left(\frac{d \ln \gamma i}{d\,T} \right)_{P,x}$$

Where the second term indicates the heat of mixing and in this study it was neglected. The final equation is therefore,

$$H_m^{L} = X_1 \left[100 + \int_{Tref}^{T} Cp_1^{L}\, dT \right] + X_2 \left[100 + \int_{Tref}^{T} Cp_2^{L}\, dT \right] \tag{24}$$

These values of saturated enthalpies for liquid and vapour, at constant pressure, were plotted against liquid and vapour compositions X, Y respectively.

Discussion

As there was no experimental data available for either DMF or for this mixture, comparison of results was not possible. The work is under progress and it is expected that even this analysis could be further improved by getting a more reasonable value

of interaction coefficient and then of δ. Secondly, the heat of mixing in the liquid phase should also be taken into account as it is quite significant specially for this mixture. It can be concluded that keeping in view the importance of this mixture for vapour absorption refrigeration system, this analysis and the properties are of great interest.

Notation

A,a,B,b,	=	Constants in Redlich–Kwong Equation
f_i	=	fugacity of pure component i at temperature and pressure of system
$\bar{f}_i$	=	fugacity of component i in mixture
H	=	molal enthalpy of mixture
ΔH_i	=	isothermal effect of pressure on the partial enthalpy or the mixture enthalpy difference $(H^P-H^O)_T$
$\bar{H}^P_{i,T}$	=	partial enthalpy of component i at temperature T and pressure P
H^O_T	=	ideal gas enthalpy of the mixture at the temperature T
H^P_T	=	enthalpy of the mixture at temperature T and pressure P
P	=	absolute pressure
R	=	universal gas constant
S	=	entropy
T	=	absolute temperature
V	=	specific volume
V_i	=	molal volume of pure i at temperature and pressure of mixture
ΔV	=	volume change of mixing per mole of mixture
X_i	=	mole fraction of component in liquid phase
Y_i	=	mole fraction of component in vapour phase
Z	=	compressibility factor
γ_i	=	activity coefficient of component i in mixture
ϕ_i	=	Fugacity coefficient of component i

Superscripts:

L	=	Liquid phase
V	=	Vapour phase

Subscripts:

C	=	Critical value
T,P,V,y,	=	values held constant during an operation

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SESSION III

Design and Construction of Cold Storages

Chairman

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Indian Institute of Technology, New Delhi

Rapporteurs

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Scientist, CFTRI, Mysore

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Survey of Cold Storages in and Around Ludhiana

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Cold storage is storage of fresh fruits and vegetables in the temperature range of – 1.1 to 10°C (30° to 50°F); storage at temperatures below –1.1°C is termed as freezer storage. Table 1 shows that the cold storage installed capacity has increased 530 times over a period of 25 years; but it is still dwarfed by the country's production of cold storage items of 20.62 million tons, excluding 56 million eggs and dairy products. The present capacity of cold storage and freezer storage in the country is sufficient for merely 14 per cent of potato production, 0.5 per cent of fruits, vegetables, eggs, etc., and 0.4 per cent of fish.

TABLE 1. *Yearly increase of cold storage capacity*

Year	Cold storage capacity in tonnes	No. of units in operation	Remarks
1947	3145	4	—
1952	59200	100	—
1955	77145	—	—
1960	305513	359	—
1965	682100	600	64% units have capacity up to 1000 tons
1966	900000	706	—
1968	1000000	937	—
1971	1506000	1310	—
1972	1679000	1390	—

In the Fourth Five Year Plan, the Central Warehousing Corporation, State Warehousing Corporations, the National Cooperative Development Corporation, and the Marine Products Corporation, had plans to setup networks of cold storage and frozen food storage facilities across the nation. All these are good auguries of the future scope of cold storage facilities in the country.

1. Cold Storage Equipment

The necessary equipment for a cold storage plant consists of a compressor, condenser, receiving tank, liquid control device, evaporator, and suitable piping and ducting. Controls for automation are also installed.

Figure 1 shows typical plan of a cold storage. The evaporator or the diffuser with fans and ducts are installed inside the specially built, insulated chambers of the

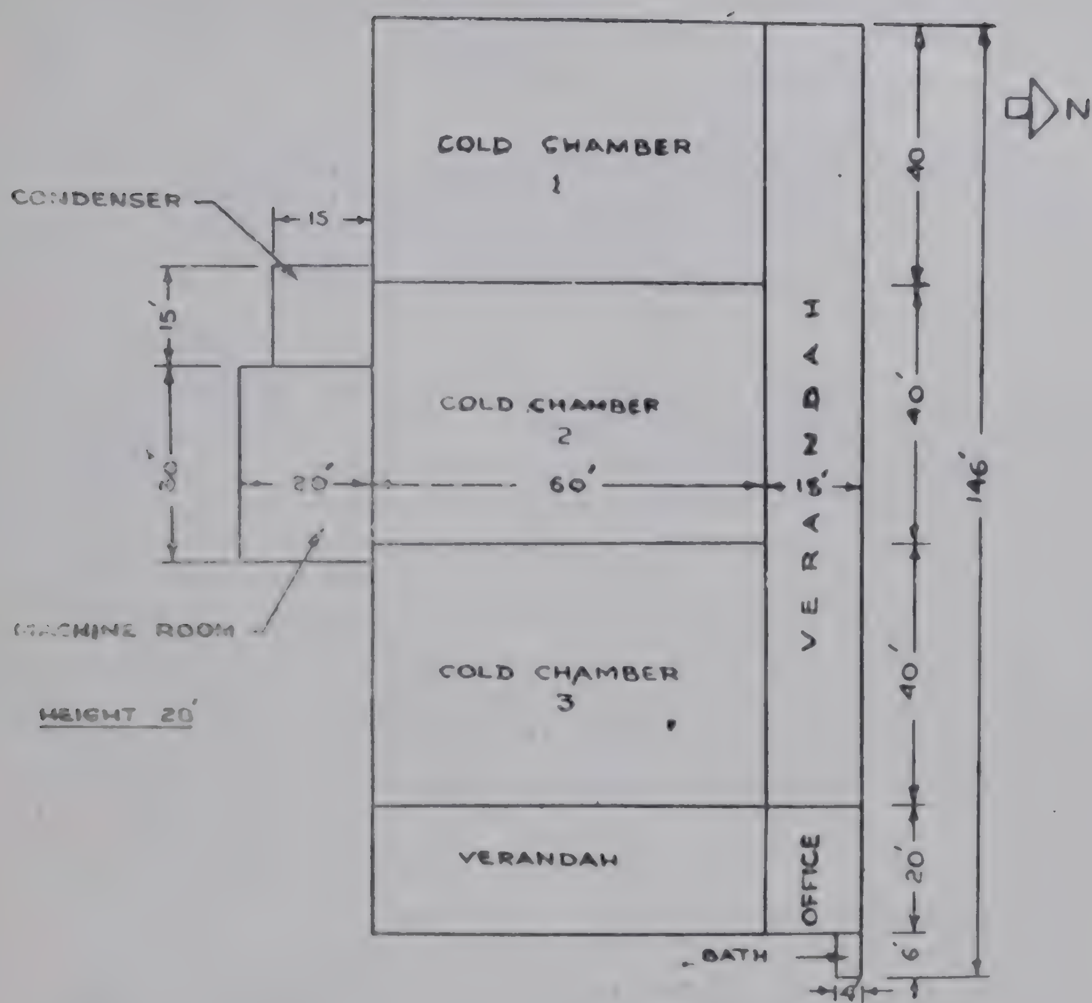

FIG 1. LAYOUT OF A COLD STORAGE

cold store meant for holding the produce. The chambers are divided further into 3 or 4 stages by wooden gratings and posts. Each stage might be 2.4 mds (8 ft) high. In this way it is possible to store potatoes at the rate of 1 ton (30 mds approximately) for 3.36 cu.m. (120 cu. ft) of space. All the other machinery is installed in the machine room located generally at the rear of the chambers. Covered verandahs which also serve as drying areas, and offices are other buildings which form a part of the complex.

2. Planning a cold storage

The planning of a cold storage facility requires : (a) preparation of a feasibility report; (b) selection of site : (c) selection of proper equipment; (d) selection of proper insulation material for the cold storage chambers ; (e) designing a suitable stacking plan ; and (f) plans for operating the cold storage year round and expansion facilities.

The feasibility report lays heavy stress on financial implications. The storages should be located either in producing or in consuming areas. They should be near

the main roads, railway stations or mandis. There should be enough area for the machine room, cold chambers, offices, product drying yards and space for smooth movement of vehicles. Electricity and water supply must be assured at the site.

The decision on equipment selection involves : (a) use of freon or ammonia as a refrigerant ; (b) low speed compressor or high speed compressor ; (c) automatic or manual control systems ; (d) cooling coils for maintaining proper relative humidity ; (e) type of condenser ; (f) use of forced air circulation or bare coils ; and (g) stand-by unit.

Selection of insulation material for cold storage chambers has an influence on the electricity bill, performance of machinery, availability of storage space and warranty against spoilage and fires, besides determining initial building costs. M.Bala Subramaniam (1) has stated that there was a net annual gain by using a polystyrene foam insulation instead of sawdust. The figures quoted show that the higher initial cost of using a polystyrene foam insulation can be wiped out within four years of operation. Manmohan Singh (2) has explained in a lucid way the selection criteria of the equipment based on the above considerations. A suitable stacking plan must allow ample space for free movement and for easy checking and removal of products. Efforts should be made to run the cold storage the whole year round, storing different products in different seasons and also making blocks of ice in the lean period.

3. Cost calculations and analysis

The overall cost includes the owning costs and operating costs. The owning costs are :

(a) capital recovery and interest ;

(b) taxes ;

(c) insurance.

The operating costs involve :

(a) energy ;

(b) operating and maintenance services ;

(c) materials and supplies.

3.1 Owning Costs

3.1.0 Capital Recovery and Interest—The capital recovery is described by amortization and depreciation. Amortization is periodic payment of money to pay back the loan. Depreciation is the "process of allocating the first cost of a capital asset over estimated useful life of asset". The useful life of different cold storage machinery and materials is given in Table 2.

TABLE 2. *Useful life or minimum depreciation period of some cold storage equipment*

Item	Years	Item	Years
Air Terminals	15	Dehumidifier	10
Diffuser			
Grills/registers			
Ceiling	20	Ducts and other sheet metal work	± life of bldg
Wall Type	20		
Window			
Plaques	20	Engines	20
Absorption type chillers	20	Fans	15
Compressors		Filters, air, oil, self-cleaning	20
Air for pneumatic controls	20		
Refrigerating	20	Dry cleanable filter	10
Condensers		Gauges	15
Double pipe	20		
Evaporation	15	Insulation, paddy husk	8
Shell and tube	20	Asbestos	15
Coolers, water tank, coil or shell and tube	20		
		Cork	20
Louvres & screens		Magnesia hot pipes	15
Copper	25	Wool felt	20
Steel	15	Thermocole	25
Manometer	15	Spray bond	15
Motors	20	Starter, electric	20
Piping, refrigerant and other	20	Switch board, electric	25
Pumps		Thermometers	15
Chilled water	20		
Condenser water	20	Tower cooling	15
Condensate	20		
Dehumidifier	22	Transformer	25
Evaporative condenser	15		
Sump	25	Turbines	
Well	25	Gas	20
		Steam	30
Receiver for refrigerant	25	Valves	
Regulator	5	Relief	20
		Automatic expansion	5
Silica gel beds	15	Water regulating solenoid	20
		Solenoid	15
Wells & well pumps	25	Building	80

Calculation of depreciation on equipment and building is shown in Table 3.

TABLE 3. *Depreciation rates calculation*

Sl. No.	Description	Life (years)	Cost (Rs.)	Rate (Rs.)
1.	Compressor	20	1,00,000	5,000
2.	Condenser	20	80,000	4,000
3.	Receiver tank	25	6,000	240
4.	Expansion valve	5	1,500	300
5.	Solenoid valve	15	4,200	280
6.	Gauges and thermometers	15	2,500	166
7.	Thermostats and cut-outs	10	1,200	120
8.	Safety relief valves	20	3,200	160
9.	Diffuser	15	75,000	5,000
10.	Piping	25	6,200	250
11.	Humidifier	10	10,000	1,000
12.	Building	80	90% of initial cost	2,040
13.	Insulation	25	1,00,000	4,000
14.	Machine foundation	80	90% of initial cost	35
15.	Stacking racks	10	25,000	2,500
16.	Tubewell pump or motor	20	2,000	100
			Total	25,191

After determination of the proper rate of depreciation and amortization period in years, the initial cost (IC) is converted into an equivalent uniform annual cost for 'n' years through the use of the capital recovery factor (CRF). The equivalent annual cost of asset is calculated by multiplying (IC) by (CRF) which is determined with the help of Table 4.

TABLE 4. *Capital recovery factors*

Years	Rate of return or Interest rate 8	10
2	.56077	.57619
4	.30192	.31547
6	.21632	.22961
8	.174011	.18744
10	.14903	.16275
12	.13270	.14676
14	.12130	.13575
16	.11298	.12782
18	.10670	.12193
20	.10185	.11746
25	.09368	.11017
30	.08883	.10608
35	.08580	.10369
40	.08386	.10226

3.1.1 *Taxes*—These vary according to the practice in vogue in the areas.

3.1.2 *Insurance*—The rates vary with the location and type of structure and nature of owner's business. Exact insurance rates should be verified from the owner's insurance company or the underwriter.

3.2 *Operating Costs*

3.2.0 *Energy*—Energy is required for: refrigerating equipment, pumping equipment, fan equipment, indoor lighting, outdoor lighting, office machinery, etc.

Heat energy is used for absorption refrigeration type. Energy loads are determined for each item and then assigned a time schedule of use. The magnitude of each scheduled load which is applicable to portions of load should be determined. The factors affecting these loads and hourly variations from maximum design values are the cold storage occupancy schedule, temperature control programme, and refrigeration equipment type and design.

The hourly load calculations are analysed to obtain simultaneous peak demand. The monthly demand peak and input energy requirements are then used with rate schedules to obtain actual cost of estimated energy requirement.

3.2.1 *Maintenance Allowance*—It includes the expenditure on the labour and spare parts and materials required for carrying out repairs; replacement of materials like refrigerant; inspection; painting and cleaning etc.

An estimate based on the early period of operation would be erroneous, as the repairs then are usually very minor. The entire period should be reckoned with to arrive at an accurate estimate. Contractual service is also included in this item.

3.2.2 *Labour for Operation*—Operators are generally charged to maintenance allowance. The operators are required to do other normal jobs, when cold storage is in operation. These charges are to be apportioned accordingly. The office staff salary is included under this head.

3.2.3 *Water Costs*—The following equation is used to estimate seasonal water usage:

$$G = B\ (TH)$$

where G = gallons of water evaporated per season,

B = Constant depending upon type of refrigeration equipment (for electric motor driven chillers B is 3.2 and for absorption chillers, B is 6.2);

TH = Tonne-Hours of refrigeration per season.

A tonne-hour of refrigeration is equal to a cooling rate of 12,000 Btu per hour. Refrigeration load is generally 900 Btu per ton of product.

3.3 *Cost Estimate*

The cost analysis is now illustrated with a typical example. Data have been analysed after conducting a survey of cold storages in Ludhiana, Jullundur and Chandigarh.

The dimensions of a typical cold storage plant are given in plan view in Figure 1. Such a plant, having 3 chambers each of 12m × 18m (40′ × 60′) and 8.4m (28′) high, can accommodate 1690 t or 50,700 mds in 5,656 cu m (2,01,600c ft.) of

space. The plant has the capacity to store 30,000 bags of potatoes at a time. The products generally stored are potatoes, and fruits like oranges, apples and '*alubukhara*'.

The storage period of potatoes is from February to October and the rate is Rs. 10.00 per bag per season. Oranges are stored from March to June and are charged Rs. 1.50 per box of 30-35 kg per month. Apples are charged similarly and are stored from July to October (one box contains 8-10 kg), *alubukhara* are stored in May and June and are charged 50 paise per box of 5-7 kg per month.

3.3.1 *Owning Costs*

A. Equipment		
1. Energy and fuel services		
(a) Electrical service entrance and distribution cost	Rs.	x
(b) Total energy plant	Rs	x
2. Refrigeration equipment		
(a) Compressors, chillers, or absorption units (4 units, 1 standby)	Rs.	1,00,000
(b) Cooling towers, condensers, well-water supplies :	Rs	1,00,000
(c) Refrigeration equipment auxiliaries (receiving tank etc.)	Rs.	6,000
3. Cooling distribution equipment		
(a) Pumps, piping, piping insulation, condensate drains, etc.	Rs.	6,200
(b) Terminal units, mixing boxes, diffusers, grillers, etc. (3 units with ducts)	Rs.	70,000
4. Air treatment and distribution system		
(a) Humidifiers, dehumidifiers, filters etc. (b) Fans, ducts, ducts' insulation dampers etc (included in 3 b) (c) Exhaust and return systems	Rs.	10 000
(5) Control systems, complete	Rs.	15,000
B. Building, construction and alteration :		
(a) Land (31,000 sq ft or 0.7 acre)	Rs.	17,000
(b) Building, 3 chambers, (7200 sq ft.)	Rs.	1,44,000
(c) Office, shed etc. (5,500 sq ft)	Rs.	55,000
(d) Compound walls	Rs.	5,000
(e) Insulation (200 cu m approximately)	Rs.	1,00,000
(f) Machinery, foundation, etc.	Rs.	3,500
C. Wiring and piping costs	Rs.	14,000
D. Installation costs (supervision)	Rs.	15,000
E. Miscellaneous	Rs.	8,000
Total Initial cost (IC)	Rs.	6,68,700

II.	Annual fixed charges		
A	Amortization period (No. of years, n)		5
B.	Interest rate (i)		10 per cent
C.	Capital recovery factor (C.R.F. from Table)		0.31547
D.	Equivalent uniform annual cost (CRF x IC)	Rs.	2,10,950
E.	Income Taxes	Rs.	22,000
F.	Property taxes (House tax inclusive)	Rs.	362 + 625
		=Rs.	987
G.	Insurance	Rs.	2,500
H.	Rent	Rs.	x
I.	Depreciation (from Table 3)	Rs.	25,191
	Total annual fixed charges (AFC)	Rs.	2,61,628

3.3.2 *Operating Costs*

I.	Annual maintenance allowance		
	1. Replacement of air or water filters	Rs.	x
	2. Contracted maintenance service	Rs.	2,000
	3. Lubricating oil and grease	Rs.	1,500
	4. General house keeping costs	Rs.	2,000
	5. Replacement of worn parts (labour & materials)	Rs.	3,000
	6. Refrigerant (50 kg @ Rs. 8/— per Kg) ammonia	Rs.	400
	Total annual Maintenance allowance	Rs.	8,900 or
		Rs.	9,000

II.	Annual energy and water costs		
	(a) Electrical energy costs (800 units approx. per day)		
	1. Chiller or compressor	Rs.	12,000
	2. Pumps: a. Chilled water; b. Condenser; c. Well-water	Rs.	2,000
	3. Fans: a. Condenser or tower; b. Inside air handling; c. Exhaust; d. Makeup air	Rs.	4,000
	4. Miscellaneous (Transformer Rent)	Rs.	288
	(b) Water		
	1. Condenser make up water; 2. Sewer charges; 3. Chemicals	Rs.	2,000
	Total annual energy and water costs	Rs.	20,288

III. Wages of Engineers and operators

Manager	1	Rs. 500	
Operators	2	Rs. 600	
Operator	1	Rs. 250	
Office assistant	1	Rs. 200	
Chowkidar	1	Rs. 125	
Sweeper	1	Rs. 50	
Mali	1	Rs. 125	
		Rs. 1850 × 12	Rs. 22,200

IV. Other Overhead Costs

Stationery		Rs. 3,000
Entertainment		Rs. 7,000
	Total	Rs. 10,000

3.3.3 *Abstract of Costs*

An abstract of the cost estimate for a cold storage is presented below :

TABLE 5. *Abstract cost estimate of a 1690 T Cold Storage*

Capital costs	Actual cost	Cost per ton Capacity
	Rs.	Rs.
Building and site	3,24,500	
Machinery	2,92,200	
Service, Installation Planning and Design Supervision Miscellaneous	52,000	
Total capital costs	6,68,700	395.50
Annual Fixed Costs		
	Rs.	Rs.
Loan amortizatioa	2,10,950	
Total depreciation	25,191	
Taxes, etc.	25,487	
Total annual fixed costs	2,61,628	154.80
Water and power	20,288	12.00
Other overhead costs		
Permanent staff	22,200	
Office, stationery, entertainment, etc.	10,000	
Total annual overhead cost	32,200	19.05
Total annual cost per ton capacity		185.85
		or 186.00

4. Net return and discussions

As calculated, the cost per ton is Rs. 186.00. The rate is high and is obviously due to amortization money. The amortization money will be paid in full in 5 years,

and the annual cost per ton will then amount to Rs. 62.00 per ton. The average lowest charge for the season is about Rs. 120.00 per ton or Rs. 4.00 per md. The annual net return thus works out to be Rs. 58.00 per ton. For a 1690 tonne cold storage the total net returns will be about Rs. 98,020.00.

In practice, the accurate net return is difficult to evaluate in advance. It depends upon the cold storage occupancy throughout the year, production of the crop, and also the convenience to the producer and consumer (which is difficult to measure).

TABLE 6. *Occupancy and annual net returns from a potato Cold Storage of 1690 tonne capacity after five years*

Sl. No.	Occupancy	Cost in Rs.	Occupancy charges in Rs.	Net return in Rs.	Remarks
1	Full or 1690 tonne	1,04,780	2,02,800	98,020	Profit
2	3/4 cap. or 1266 tonne	1,04,780	1,51,920	47,140	Profit
3	1/2 cap. or 845 tonne	1,04,780	1,01,400	3,380	Loss

There are seasons when rates go high, thereby increasing the profitability. Sometimes the occupancy may fall below ½ and thus result in losses. The minimum level of stored potato tonnage should be 875 for no profit and no loss. The produce stored also gets damaged sometimes due to bad management and power failure.

It is recommended, therefore, to store different types of products and ensure year round full occupancy. Making ice blocks is one activity which helps in lean periods.

Summary

Cold storage capacity in the country has increased lately. It is still insufficient to meet the production of perishable products. It is important to give careful attention to the selection of proper equipment and insulation material for the cold storage construction. A typical cost analysis has shown that it is a profitable proposition if the occupancy remains more than half of built up capacity for the year.

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About Jacketed Cold Stores

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Introduction

The jacket system of cold storage has now reached a state of commercial development. Jacketed storages have now been built in a number of European countries including the U. S. S. R., and in Canada (Table 1). In the United States, the jacket system is used in many mechanically cooled railway refrigerator cars. In our country the construction of jacketed stores have not been accorded the same priority as the conventional cold stores.

The jacketed room or envelope system has many advantages over the conventional cold storages in that it provides ideal conditions for the storage of frozen

TABLE 1. *Commercial use of the jacket system for cold storage*

Country	Number	Total capacity (m^3)	Use	Remarks
Austria	1	8,000	Fresh and frozen food	Jackets enclose three blocks each with five rooms.
Canada	2	6,000	Frozen foods	Fully jacketed forced air circulation.
	2	3,000	Apples	Fully jacketed, forced air circulation, controlled atmosphere.
France	2	5,000	Multipurpose.	By-pass system used for direct cooling.
Germany (Federal Republic)	No data available.	No data available.	Nursery stocks.	Fully jacketed, forced air circulation.
Japan	3	20,000	Frozen foods	Fully jacketed, forced air circulation.
	Many	17,000	Fruits & vegetables	Partially jacketed, forced air circulation
Netherlands	About 50	10,000 (estimated)	Mainly fruits & vegetables.	Mainly partial jacketing.
Switzerland	1	5,000	Fresh & frozen foods.	Exterior walls & ceilings jacketed forced circulation.
U.S.S.R.	1	120,000 (estimated)	About 75% for frozen foods.	Each storey separately jacketed, gravity air circulation.
	Grand total	194,000		

foods; uniform temperature, high and uniform relative humidity, and low and even air velocity. The jacket system also provides a solution for the problem caused by condensation of moisture in the insulation of refrigerated spaces, a problem often aggravated in fruit and vegetable storages by reversal of the vapour pressure gradient. The deterioration of insulation as a result of condensation is particularly important where internal or external temperatures vary and where seams and joints are subject to flexing, as in a refrigerated transport vehicle. The jacket system is an obvious solution for this problem of controlled atmosphere storage where a highly vapour-resistant lining is required on the cold side of the insulation. Among other advantages offered by the jacket system are greater flexibility in selection of insulation and evaporator, reduction or elimination of the defrosting problem and more efficient evaporator operation. Increased cost of construction appears to be the main disadvantage of the system, with the jacketed load bearing floor accounting for much of the extra cost. The larger fan used in the jacket system increases the power required by about 10% but this may be more than offset by the increased efficiency of heat transfer.

The object of this paper is to discuss the various factors pertaining to the different aspects of jacketed cold stores including its principles of design and operation.

The air jacket

The loss of weight of the food stuffs stored in a cold store is in direct relation to the amount of heat penetrating into the cold rooms. Along with the loss of weight (shrinkage) of the food stuffs the quality is also impaired due to the dehydration of the surface layer of the products. This in turn, results in a reduction of the nutritive value of the food stuff as well as a deterioration of its appearance as many products (eg. eggs, fruits, frozen meat etc.) are stored at cold stores without a vapour proof packing.

It has been shown, that each 1000 Kcal. of heat penetrating into the cold room causes a 150g. loss of weight of the stored products. These factors prompted the construction of an air jacket which would absorb the heat gain from the out side before the latter penetrated into the cold rooms, while the internal heat gains (product load, operation loads etc.) would be absorbed by the coils and air coolers installed in the cold rooms.

Developments of the jacket system

In order to reduce the weight loss of produce in cold stores, a more economical method was first suggested in Canada by Huntsman in 1931. The proposal was to surround the cold storage space with an air-tight jacket or envelope containing the cooling surfaces. The insulation of the store is placed on the outside of the jacket and air is circulated within it to maintain a very uniform temperature in the lining of the storage space. Under steady conditions, the dew-point of the air within the storage space is given by the lowest temperature of the lining (Fig. 1). By this method, very high humidities may be maintained inside the store but the

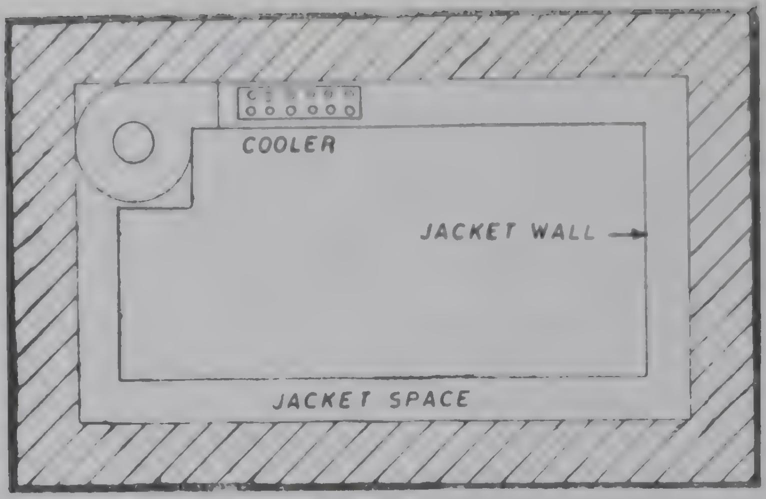

FIG. 1. Diagramatic sketch jacketed store

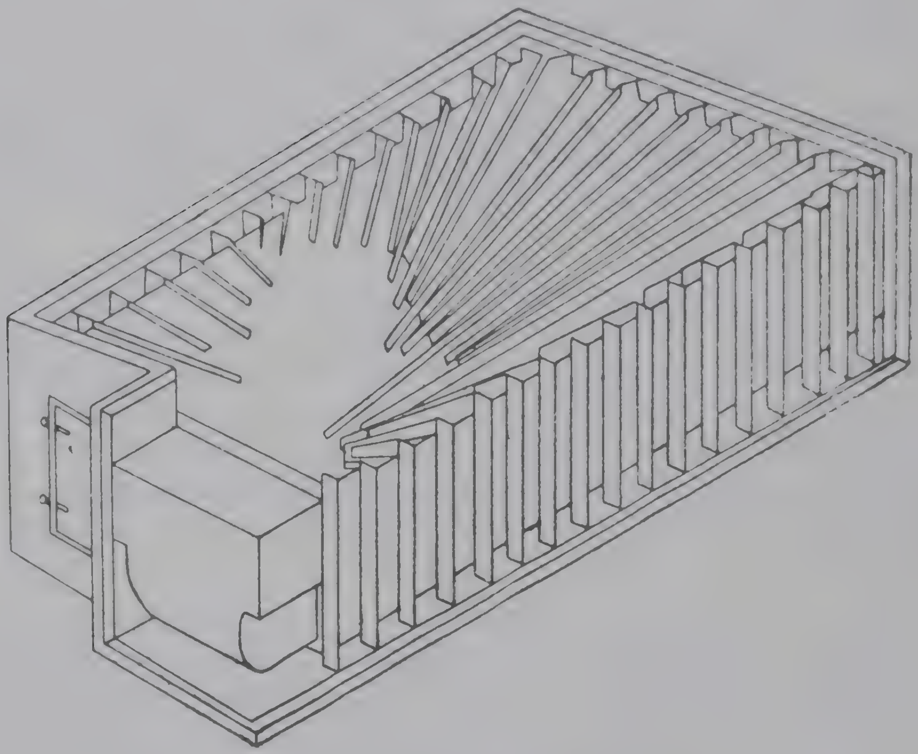
FIG. 2. Canadian type jacketed store

system has not won general acceptance as only frozen goods in sealed consumer packs can be stored.

Fig. 2 shows a sketch of a Canadian design employing a unit cooler on an end wall and fan like distribution and collection ducts under floor and ceiling. The flow in the three remaining walls is in parallel and the depth of ducting in these walls is much reduced in relation to the floor and ceiling ducting in order to provide even distribution. The complication involved are easy to see and even the possibility of reduced insulation cost has not been sufficiently attractive to encourage the construction of this type of store.

The Torry ducting system illustrated in Fig. 3 overcomes many of these difficulties. In the Torry system the jacket spaces in floor and ceiling are each divided into two parts by means of diagonal sealing strips. The floor space and the ceiling space are divided along the opposite diagonals of the rectangular store plan. The air flow is illustrated in Fig. 3. This simplifies the distribution problem because there is little chance of the air finding a short circuit or low resistance path round the system. It is a feature of the Torry ducting system that heat is extracted from the jacket air at two points in the circuit thus maintaining it at a more uniform temperature than would be possible with a single cooler.

An important consideration in the design of jacket stores is that there should be a good vapour seal between room and jacket and between outside air and jacket.

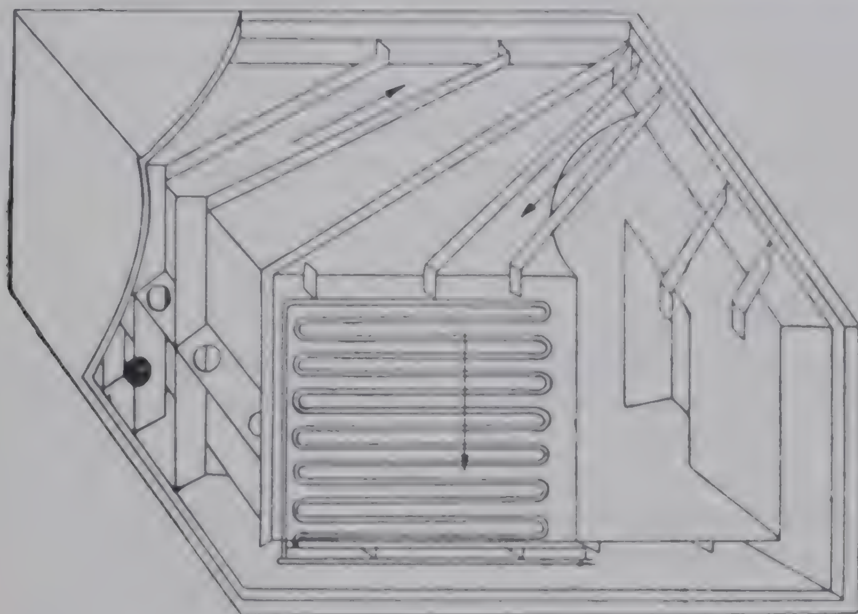

FIG 3. Torry system of ducting for jacketed store

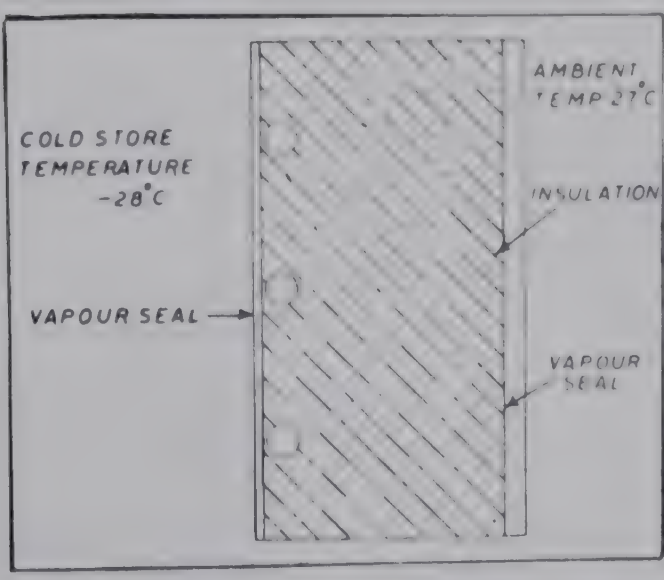

FIG. 4. Wall section conduction jacketed store

This problem is largely solved by the use of modern foa ned-in-place insulation which is used to form a continuous unbroken seal. It is felt that a considerable proportion of the increased construction costs resulting from the use of a sealed jacket could be met by the reduction in the amount of pipe work required, by reduced produce weight losses and by cutting down the thickness of insulation to a value which is economically more justifiable.

The use of foamed-in-place insulation will probably solve the sealing problem in jacket cold stores. In the new construction the cooling is done by pipe grids which are contained just within the inner surface of the foamed-in-place insulation as in Fig. 4.

The cooling grids are placed in walls, ceiling and floor of the store to intercept the heat which leaks in through the insulation. A uniform temperarure in the lining of the cold store is maintained by using 16 gauge aluminium sheet to form the inner boundary of the insulated wall. As the conductivity of the aluminium is over 5,000 times of the insulated wall it is readily seen that very uniform temperatures can be maintained in the cold store lining.

Principles of operation

In the jacket system, heat leaking through the room insulation is absorbed by a refrigerated air stream flowing around the entire periphery of the room but separated from the atmosphere of the room itself, or storage space, by a water vapour proof jacket. (Inner jacket wall, Fig. 5). In the system shown, air cooled by the evaporator at one end wall space, enters the ceiling air space and flows down through the wall spaces on the other three sides into the floor space and back to the evaporator. It can be seen that relative humidity in the storage space is not directly dependent on external heat load or evaporator operation, that there will be no sharp temperature gradients near walls, and that vigorous air movement in the room is not required.

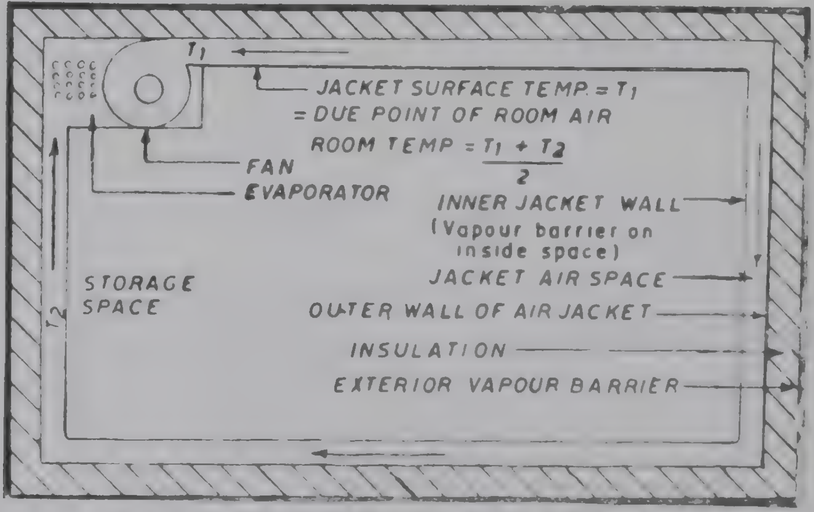

FIG. 5. Schematic sketch of jacketed cold room

The relation between heat absorbed and the amount and temperature of the air circulating through the jacket space is given by

$$Q = W\,C(T_2 - T_1)$$

Where Q = heat absorbed per unit time

W = weight of air circulated per unit time.

C = specific heat of air.

T_1 = temperature of air leaving evaporator to enter jacket

T_2 = temperature of air entering evaporator from jacket.

Ideally, with uniform air circulation in the jacket space and no heat load in the room itself, the average room temperatnre will be $\frac{T_1 + T_2}{2}$ while the dew-point will be equal to the coldest surface temperature T_1. For any heat load Q, T_1 and T_2 depend directly on W. Hence relative humidity, a function of T_1 and T_2 depends on W. In practice, temperature and humidity conditions in the room are also affected by spatial variations in air flow, temperature in the jacket and heat loads in the room. Some heat always flows into the room through the structural members of the jacket in addition to that coming from lights, electrical equipment, and air changes during door openings. Uneven flow distribution in the jacket increases the amount of heat entering the room through the jacket. These internal heat loads raise the room temperature above $\frac{T_1 + T_2}{2}$ and hence lower relative humidity.

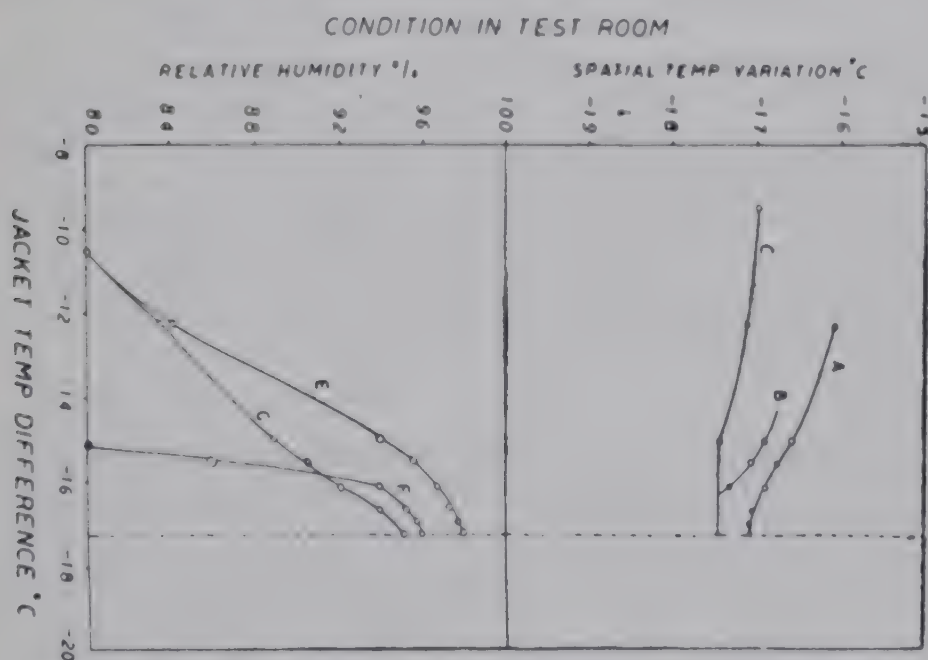

FIG. 6. Effect of condition in test room on jacket temperature difference

A : Bottom-to-top circulation in jacket ;
B : Low air flow resistance ;
C : Increased resistance ;
D : Top-to-bottom circulation in jacket ;
E : Increased Resistance
F : Top-to-bottom circulation in jacket low air flow resistance
G : Bottom-to-top circulation in jacket

It has been shown by a test, conducted in an empty jacketed test room (6x3x3m) at -10°C (1) that the main factors affecting relative humidity and spatial variation in temperature were direction of air movement in the jacket(top-to-bottom or bottom-to-top), rate of air flow in the jacket and air flow resistance in the jacket (Fig. 6).

Heat-transfer associated in jacketed cold stores

Respiration heat of living produce has to be transferred to the walls of the jacket. However, this is accompanied by condensation of moisture which can be very inconvenient, especially in case of condensation on the ceiling. Furthermore heat generation causes an unequal temperature distribution in the store. Hence temperature distribution in the jacketed stores should be studied and heat transfer to the walls of the jacket must be arranged in such a way as to prevent moistening of the ceiling.

In order to decrease the condensation, lowering of temperature difference at the ceiling has been attempted by the various following means :

(a) Cooling the ceiling as little as possible ; (b) Constructing the connection between walls and ceiling of the jacket of a poor heat conducting material ; (c) insulating ceiling at the outside ; (d) insulating ceiling at the inside, and (e) constructing ceiling of a material with a small radiation number. It has been seen that the arrangements mentioned under (b) and (e) were unsatisfactory. Insulatlng the ceiling on the inside gave good results, but is difficult to realize.

By reducing heat transfer at the jacket to zero, condensation at all surfaces can be prevented. This can be reached by using, inside of the jacketed store, as air cooler is capable of compensating respiration heat. Air flow through this cooler should be contrary to the convective flow. The cooling unit in the jacket space should be just large enough to remove field heat of the produce and heat transmitted through the insulation. It is desirable that the temperature of the inside cooling unit must be equal to that of tne jacket surface. Constructing a store in such a way usually results in fully dry ceiling and dry walls.

Design principles

The most important element in the design of a jacketed room is probably the air jacket. Ideally, air flow in the jacket should be in proportion to the heat to be absorbed. For top-to-bottom or bottom-to-top circulation a practical design would be one in which the flow resistance in the wall spaces is uniform all around the room. In such a system the air would enter the floor or ceiling air space, get distributed uniformly in the wall spaces (because of the relatively high flow resistance of the wall spaces) flow vertically in the wall spaces (structural members acting as guides) and return to the evaporator from the opposite ceiling or floor air space.

The depth or thickness of the wall, floor, and ceiling air spaces depend on the rate of air circulation and on the flow resistance required. Flow resistance in the walls should be sufficient to ensure uniform distribution and efficient operation. It is observed that in a test room (insulated with 20cm. of rock wool) the minimum

pressure loss that gave good results was equivalent to 0.25mm of water. Since the flow resistance required to ensure efficient operation of the jacket is negligible compared to the pressure loss normally occuring in fan and evaporator systems, a large safety factor may be allowed. In general, the wall air space may be between 1 cm. and 5cm. in thickness, depending on rate of air flow and coefficient of friction of the material facing the two sides of the air space. The floor and ceiling spaces should be thick enough to limit flow resistance in them to a small proportion of that in the walls.

The jacket may be designed in other ways, but this information on top-to-bottom and bottom-to-top circulation should indicate the problems to be considered.

The jacket wall should have some thermal resistance and inertia to reduce the effect of temperature vibrations in the jacket air.

The jacket system allows a relatively wide choice in evaporators and exterior insulation. The need for evaporator defrosting equipment depends on the quality of the exterior and jacket wall vapour barriers since only moisture leaking through these barriers will accumulate on the evaporator. The circulating dry air should, of course, keep the insulation dry.

Where a room is to be used for freezing, or where fast cooling is required, the system should be designed so that the jacket can be by-passed, and air circulated directly through the room from evaporator.

Applications

Many specialists considered previously that the jacket presented interest only for rooms storing frozen meat and fish. However, further investigation of this problem illustrated the expediency of the jacket for chilled egg and fruit storage rooms.

It has been shown by the operation of the warehouse that the jacket permits prolonging the storage time for eggs by 1.5 to 2 months, thereby ensuring a more uniform supply of eggs to the population during the year.

Investigations and observations carried out by Scientific Research Institute of the Refrigerating Industry of the USSR on the operation of a 500 m^3 control chamber at -18°C indicate the following:

1. With automatic control of the equipment a stable temperature was maintained in the room and the temperature deviations in various parts of the room did not exceed —0.5°C.

2. The relative humidity of the air averaged 97% during the entire period of the room operation.

3. The weight losses of frozen meat (327 tons of meat were stored in the room) during the storage period for one year amounted to 0.78% for Ist grade beef, 1.24% for 2nd grade beef and 0.78% for fiirst grade mutton against the weight losses in conventional cold storage upto the extent of 1.67%, 2.18%and 2.02% respectively.

Fresh fruit and vegetable storage

The main problems that must be considered in adapting the jacket system to fresh fruit and vegetable storage appear to be :

(i) Pre-cooling (too slow), (ii) Heat of respiration (temperature and relative humidity gradients). (iii) condensation (vapour pressure of stored produce essentially equal to that of water at the same temperature). (iv) Growth of microorganisms (accelerated by high relative humidity and low air velocity). (v) Physiological effects of high relative humidity (changes in flavour, texture and firmness).

Storage tests on celery and apples (5) have shown that, some fruits and vegetables benefit from the high relative humidity, uniform temperature and low, uniform air velocity that can be maintained in a jacketed storage. The results of the tests have also indicated that high relative humidity, uniform temperature, and low, uniform air velocity can be maintained in small fruit and vegetable storage without difficulty; but it would appear that special provision must be made for air circulation through the load in larger rooms.

Examination of apples from the jacketed room and from comparable controlled atmosphere (5% carbon dioxide, 3% oxygen) and conventional rooms after five months storage indicated that the jacketed room had appreciable advantage (mainly because of a reduction in core flush) over the conventional room.

Conclusions and some suggestions for future work

Many of the jacketed rooms that have been built represent a compromise of the basic principles involved, either in the interest of reducing capital cost, or in the case of fruit and vegetable rooms, to minimize spatial temperature difference and condesation within the room. Consistent information is needed to asses the value of these departures from basic design.

Although there are other methods for maintaining high relative humidity in cold storage rooms, and other ways of attacking insulation problems none of them, appears to combine the advantages of the jacket system.

The information presented here indicates that some fruits and vegetables benefit from the high relative humidity, uniform temperature, and low, uniform air velocity that can be maintained in a jacketed storage. More information is required on the physiological effects of these conditions on other fruits and vegetables, since some do not benefit from high relative humidity. More information is also required on the effect of high relative humidity and low air velocity on the microorganisms encountered in the storage of fruits and vegetables.

Most information on the cost of construction of jacketed rooms compared with cost of construction of conventional rooms is sketchy, as the basis on which the estimates are made is not stated. More detailed information on costs is needed.

Finally, there seems to be little doubt that the jacket system offers a solution for the wet insulation problem, although conclusive proof will require observation of a number of rooms over a long period of time.

Studies should be made on jacket systems, the conception of which has a direct bearing on the work of the architect or of the engineer. Here are the main

points to be examined. Type of air circulation within large warehouse; air circulation within the jacket, and in the room; comparison between forced and natural convection of air within the jacket; influence of the size of the cold rooms, examination of the problem of rooms with a partial jacket; study of cooling in rooms provided with jacket system, and research to be undertaken on air velocities. All these studies should be based on experimental data.

Summary

The idea of air-jacketing a cold storage room appears to have been first suggested by Huntsman as a means of reducing surface desiccation of frozen fish and was later studied by others. Most of the earlier reports published on these studies indicate advantages for the jacket system compared with conventional direct cooling both from the point of view of product storage conditions and the durability and maintenance of the building. The adverse comments have been only on the increased initial capital cost, and in the case of fruit and vegetable storage, on problems that may be caused by temperature differences or moisture condensation on room or product surfaces.

Despite the information available on design and operation and the favourable reports on performance that have been published, little commercial use of the system has been made.

In the present paper an attempt has heen made to discuss the various factors pertaining to the different aspects of the jacketed cold stores, to outline the basis of design and operation of jacketed system and to present some suggestions for further work.

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Refrigeration Requirement for A Dairy Plant

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Introduction

The use of refrigeration is indispensable for the production, storage and distribution of dairy products like milk, ice-cream, butter and cheese. In modern practice milk from the milking animal flows into the bulk milk cooler where the milk is cooled to about 10 to 15°C. From these coolers, the milk is hauled by an insulated tank truck to the dairy where it is pasteurised and cooled to 2° to 4°C in a heat exchanger. To preserve the quality of milk, refrigeration is required at many later steps like distribution of milk.

In making ice-cream pasteurised ingredients are thoroughly mixed and pre-cooled to 3 to 4.5°C before freezing. The ice-cream mix is cooled to —4 to —6°C at which stage the mix is stiff but remains fluid enough of flow into containers. These containers are then removed to the hardening room at —6 to —12°C. In the manufacture and storage of butter and cheese also refrigeration is required to maintain the quality and lengthen the shelf life of the product.

Estimation of Refrigeration Load

When the process schedule has been planned, the requirements for heat, refrigeration, electricity and other services can be calculated with a certain degree of accuracy. The process schedule for a multipurpose dairy plant handling 60,000—80,000 litres of milk/day has been shown in Figure 1.

Refrigeration is required for:

i) Pre-cooling of raw milk.
ii) Final cooling of pasteurised milk
iii) Cooling of cold store
iv) Ice-cream freezing
v) Ice-cream storage
vi) Butter storage
vii) Cheese storage

The refrigeration requirement for process work and storage throughout the working day should be compiled from the data established in the process schedule. The requirements of each process equipment can be calculated from the rate of product flow and the temperature change required. It is convenient to separate the direct and indirect refrigeration. Direct refrigeration includes all the applications where the refrigerant absorbs heat directly as in cold storage and ice-cream freezers.

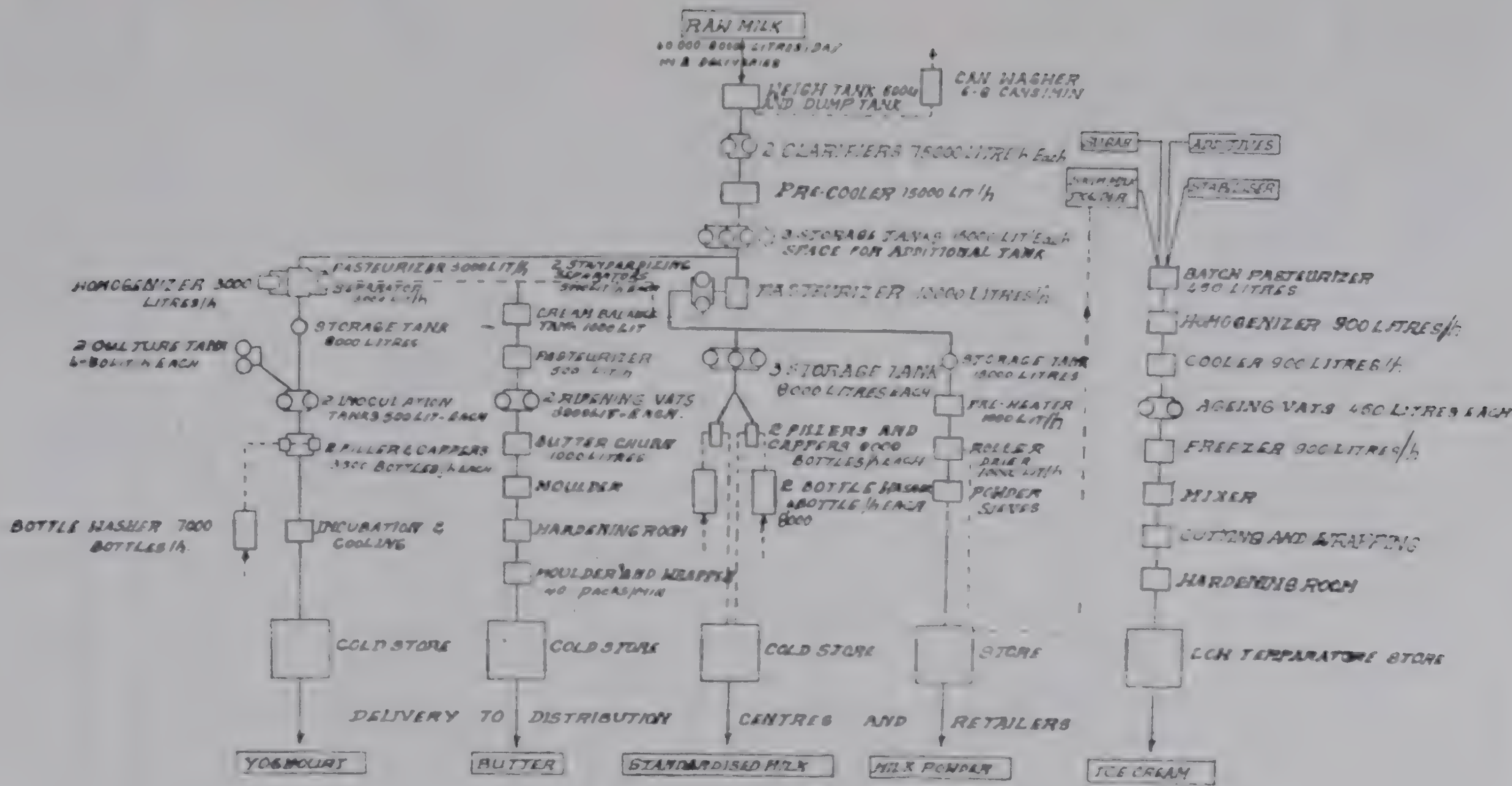

FIG. 1. Process Schedule for a Multipurpose Plant

Indirect refrigeration involves a secondary cooling medium like brine or chilled water.

The refrigeration load of cold stores is due to the heat leakage through the walls, ceiling and floor of the structure, frequent opening of the cold store during working and from crates of bottled milk. Reduction of the first two is obtained by good insulation and good management. The third is linked with the previous treatment and the temperature of bottled milk. It is usual to circulate air in the cold storage at about 60 changes per hour to give desired cooling with sufficient clearence between crates of bottled milk. Refrigeration compressors should be standardised as far as possible and interconnected with a stand-by machine to provide for emergencies and overhaul. Refrigeration load is highest during the cooling of pasteurised products.

The farmer brings his milk to the collecting centres immediately after milking which is cooled rapidly to 4°C and despatched in tankers to the milk plant. Assuming the temperature of the incoming milk to be 30°C, the refrigeration requirement for cooling this milk upto 4°C is as follows :

Heat to be removed from Milk/hour = mst.

Where m = Mass of milk to be handled per hour

s = Specific heat of milk

t = Range of desired cooling

The next process in the schedule is pasteuritation. In modern practice this is immediately followed by chilling. Therefore in calculating the refrigeration load, the flow temperatures in the various pasteurisation sections have to be taken into consideration.

t_1 = inlet temperature of raw milk

t_2 = temperature at the inlet of heating section

t_3 = pasteurisation temperature

t_4 = temperature at the inlet of chilling section

t_5 = temperature of the milk at the outlet of chilling section.

For maximum efficiency and economy the outgoing pasteurised milk is used for heating the incoming raw milk in the regenerating section. Temperature t_1, t_3 and t_5 are normally fixed.

$$\text{Regeneration efficiency } \eta_R = \frac{t_2 - t_1}{t_3 - t_1} \qquad \ldots\ldots\ldots\ldots \quad (1)$$

$$\text{Also } \eta_R = \frac{t_3 - t_4}{t_3 - t_1} \qquad \ldots\ldots\ldots\ldots \quad (2)$$

Generally the regeneration efficiency must be between 80 to 90 per cent.

The cooling of the pasteurised milk in the chilling section from t_4 to t_5 is done by chilled water. By using equation number (2) the value of t_4 can be calculated.

$$\text{Now } Q = ms\,(t_4 - t_5) \qquad \ldots\ldots\ldots\ldots \quad (3)$$

Where Q = Rate of heat transfer in chilling section.

m = Mass flow rate of milk.

s = Specific heat of milk

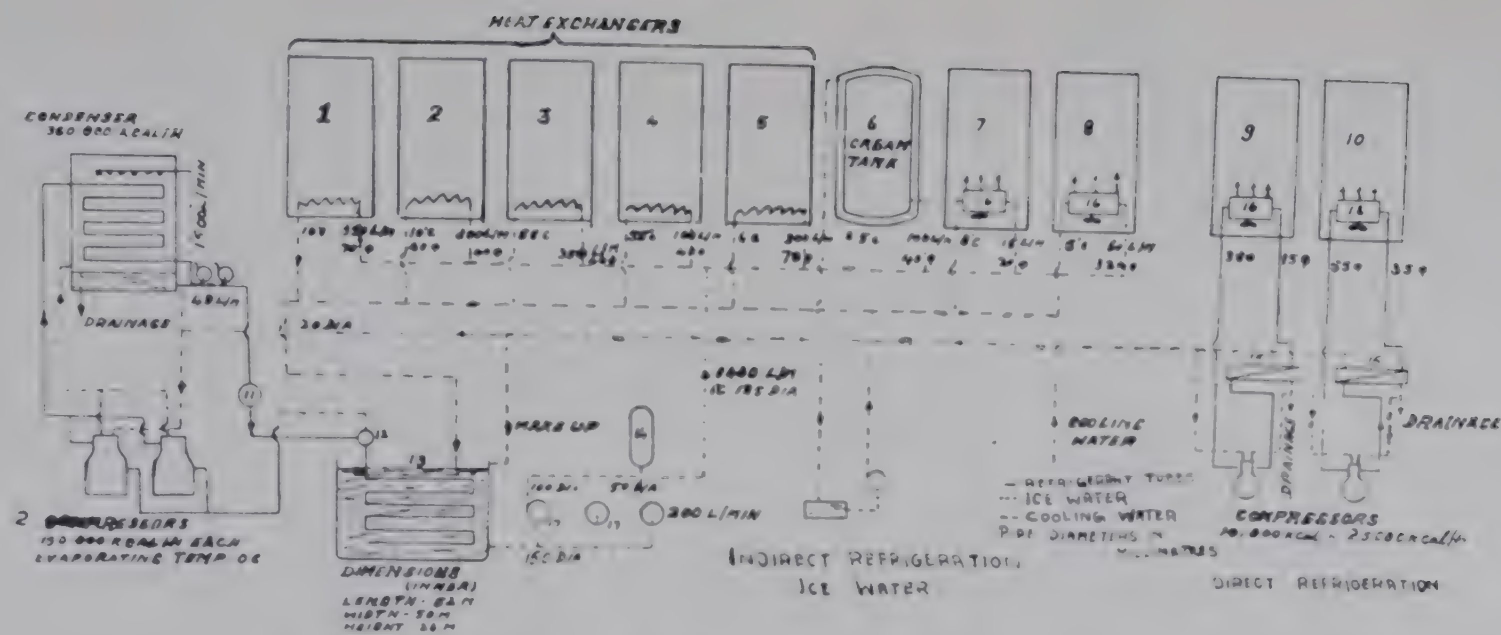

FIG. 2. Chilled Water Load Requirements

1. Pre-cooling Raw milk 800 Mcal/Day
2. Pre-cooling raw cream 33 Mcal/Day
3. Pasteurized milk 495 Mcal/Day
4. Yoghurt 63 Mcal/day
5. Butter wash water 20 Mcal/Day
6. Cream for butter making 44 Mcal/Day
7. Butter wrapping room 35 Mcal/Day
8. Cheese storages 250 Mcal/Day
9. Butter storage 2°—4°C
10. Milk storage 2°—4°C
11. Receiver
12. Surge Drum
13. Ice water basin with Evaporator
14. Pressure regulator 2000 L
15. Condenser
16. Air Cooler
17. Ice water pumps 2.700 L/Min 3 kg/cm²

This amount of heat Q is to be taken away by chilled water. By energy balance, assuming no heat losses, Heat gained by chilled water = Rate of heat transfer in chilling section.

$$W_{sw} (t_{c2} - t_{c1}) = ms(t_4 - t_5)$$

$$\text{or } W = \frac{ms (t_4 - t_5)}{S_w(t_{c2} - t_{c1})} \ldots\ldots\ldots\ldots \qquad (4)$$

W = Mass flow rate of chilled water.

S_w = Specific heat of chilled water.

t_{c2} = outlet temperature of chilled water.

t_{c1} = inlet temperature of chilled water.

In this way, the chilled water requirement for chilling of milk after pasteurisation can be estimated. In a similar manner the refrigeration load for other processes can be determined. The chilled water requirement for the composite dairy is shown in Fig. 2. The approximate requirement of refrigeration are given in the table.

1.	Cooling of milk pasteurised in H.T.S.T. plant.	2 B.T.U./Lit. of milk/°F.
2.	Bottle cold store (4°C or 40°F)	4 B.T.U./Lit. of milk. 100 BTU/sq. ft.
3.	Preparation of ice-cream for 100 lit. of ice-cream (50 Lit. of Milk).	1/3 ton.
4.	Ice-cream storing (room cooling)	120 B.T.U./sq. ft.
5.	Ice-cream hardening.	60 B.T.U./lb. of ice-cream.
6.	Butter making and storing	2 tons/ton of butter manufactured.

The refrigeration load throughout the day is not uniform in a dairy, therefore, generally the ice-bank system is used. To ensure better efficiency in a refrigeration plant, the suction and discharge pressures, temperature, condenser water, refrigerant flow etc. are controlled by automatic controllers. The controllers serve the dual function of maintaining the desired operating conditions and ensuring safety of the equipment.

Refrigeration plays a very important role in dairy industry. The demand for refrigeration for cooling, freezing, transportation and storage is increasing rapidly. Dairy management is therefore highly conscious of its responsibility towards satisfactory care of refrigeration equipments.

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Design of Refrigerated Sea Water Plant for Preservation of Fish

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Preservation of fish in refrigerated seawater has been widely employed in many of the developed countries. But the use of such method has not become popular in India, probably due to the use of only small fishing vessels. In order to study the feasibility of employing this process in Indian fishing vessels, a pilot model has been designed and developed. This paper deals with the details of the above plant.

Introduction

It has been established that lowering the temperature of fish retards the activity of enzymes and bacteria which bring about spoilage. For short time storage, upto a maximum of about seven days, fish can be kept in fresh condition by chilling immediately after catch with equal quantity of ice and subsequent replenishment. This method has been accepted in the Indian fishing industry. Similar results can be achieved when the fish is stored in refrigerated sea water (R.S.W.) maintained at 0 to —1°C. The fish is cooled much more rapidly and efficiently than in ice storage. Another important advantage in using R.S.W. storage is that fish held in this medium have buoyancy almost equal to their weight and hence do not get pressed or crushed whatever be the depth filled. In ice storage if the depth of fish and ice stored in one container exceeds about half to one meter, the bottom layers of fish get pitted and more often crushed. There is better control of temperature in R.S.W. storage which is maintained generally at —1.1°C (30°F) whereas in ice storage it is very difficult to bring the temperature below 1.5 to 2°C and uniform temperature conditions throughout the material are seldom obtained. R.S.W. storage eliminates the difficult task of icing and hence there is considerable saving of labour and ice storage space on board fishing vessels. Moreover, there need not be any fear of the ice getting exhausted and hence the fishing trip can be prolonged without the catch getting spoiled.

Basis of Design

The plant consisted of three separate refrigerated Seawater storage tanks maintained at —1.1°C (30°F). The storage tanks were chosen in such a way that in each tank a maximum of 150 kg. of fish could be stored. Even when 80% by weight of the seawater is displaced by fish, adequate passage still remains between the fish for circulating seawater for cooling.

The size of the storage tank was 61 cm. × 61 cm. × 76 cm. height. Excluding head space of 15 cm. and another 10 cm. below the perforated bottom, the actual volume available for fish is 0.1898 cu.m. The weight of sea water between the fish and below the perforated bottom was about 75 kg. Thus the cooling load on the Refrigeration machinery for one storage tank is that needed to cool 150 kg. of fresh fish from an initial temperature of say 32.2°C to —1.1°C within one hour by submerging it under flowing chilled water. It is assumed that the temperature of chilled seawater after cooling fish would be increased by about 2°C. Thus the heat load calculations are based upon cooling 150 kg. of fresh fish from 32.2°C to —1.1°C and to cool 75 kg. of sea water from 1.1°C to —1.1°C in one hour for each tank. Assuming the specific heat of fish to be 0.75 K cal/kg/°C, the total refrigeration load for removing heat from the three tanks will be 11,958.75 K cal/hr.

In order to satisfy the above conditions, a low pressure Freon–12 water cooled compressor unit was selected which is a 3 cylinder reciprocating compressor,

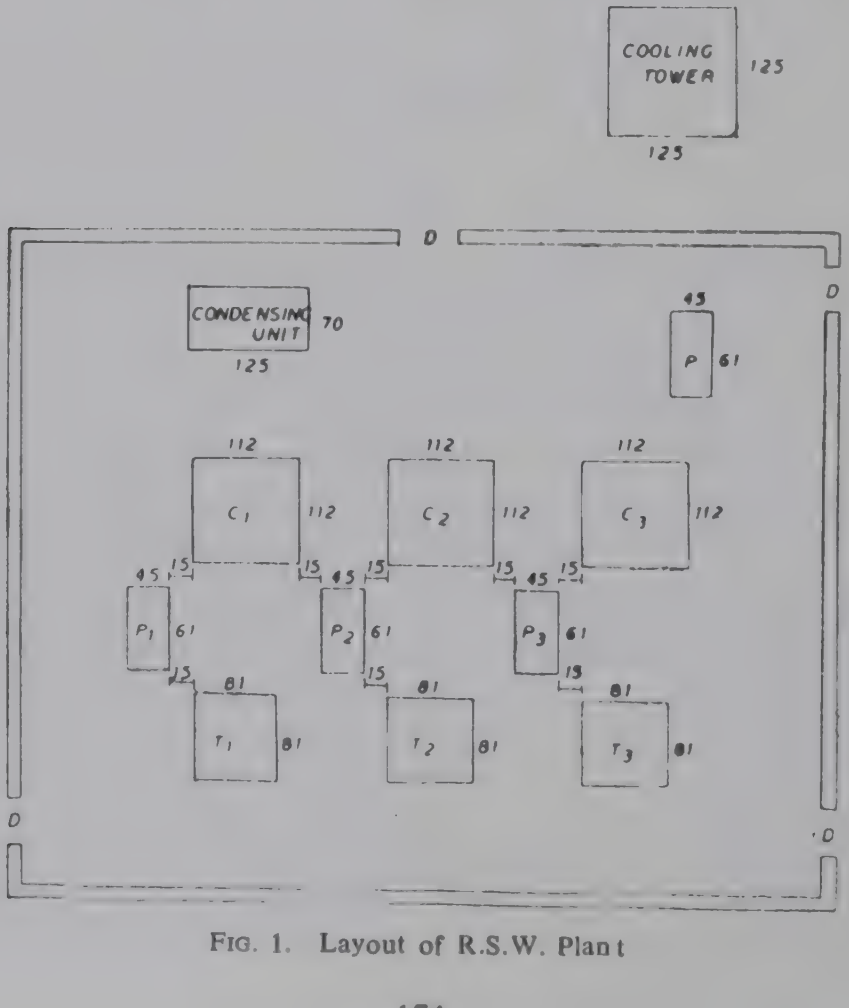

FIG. 1. Layout of R.S.W. Plan t

shell and tube water cooled condenser and receiver having capacity of 15,000 K cal/ hr. at 40°C condensing temperature and —9°C suction temperature.

The arrangements of the various components of the RSW Plant areas shown in Figures 1 T1, T2 and T3 are the three storage tanks made out of 18 gauge stainless steel and exterior constructed out of 16 gauge ms sheet. The tanks are insulated with 100 mm thick thermocole and provided with insulated doors at the top. C1, C2 and C3 are the corresponding chilling tanks also made out of 18 gauge stainless steel interior and 16 gauge ms plate outside with 100 mm thermocole insulation. Evaporator coils are made out of 16 mm cupro-nickel tubing. P1, P2 and P3 are the circulating pumps. In order to re-use the cooling water, a natural draught forced spray atmospheric type cooling tower also has been provided and the water is recirculated by using pump P to cool the compressor.

Acknowledgement

The author wishes to record his gratefulness to late Dr. V. K. Pillai, former Director of the Institute for his interest and encouragement during the course of the investigation. Thanks are also due to M/s. Frick India Ltd., New Delhi for their sincere co-operation in supplying the equipments and installing it as per our requirements. The author is indebted to the Scientists and workshop staff for their sincere co-operation without which this work could not have been completed successfully. He is thankful to Shri M. R. Nair, for carefully going through the paper and for the valuable suggestions. The author wishes to express his gratitude to Dr. S. Z. Qasim, Director of the Institute for permission to present this paper at the symposium.

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Vapour Problems in Refrigeration Insulation And Their Remedies

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A knowledge of mechanism of water vapour migration through refrigeration insulation, its effect on the performance of insulation and methods of controlling the rate of water vapour migration is necessary for proper selection and design of thickness of refrigeration insulation.

Water Vapour Migration Methods

The amount of water vapour present depends upon the relative humidity and the temperature of the ambient air; whenever the temperature at any point on or

within the insulation becomes lower than the dew point of the air on the warmer side, condensation tends to occur at that point, provided that the water vapour can migrate through the insulation from the warmer side. Vapour can also migrate through the insulation due to the hygroscopic nature of the material of insulation. Another possibility for entry of water vapour into the warm side of insulation and its transport to cold surface is by air leakage.

The flow rate of vapour migration by diffusion is usually calculated on the basis of a simple equation based on Fick's Law.

$$W = -\mu dp/dx$$

Where W = flow rate of water vapour

μ = coefficient of vapour permeability

$\frac{dp}{dx}$ = vapour pressure gradient

The permeability coefficient can be established from suitable tests

The value of μ is reasonably constant for insulations in low or moderate relative humidities.

Effects of Water Vapour Diffusion on the Insulation are :

1. Corrosion of metal work.
2. Odour and health hazards during insulation
3. Decrease of structural strength of installation
4. Increase of running and power costs
5. Poor resistance to damage and reduced service life.

Methods of controlling the flow of water vapour

Prevention and control of condensation and freezing of water vapour within the refrigeration insulation has been designed by the outstanding experiments of Row-Wey in 1937 with vapour barriers. Vapour barriers offer resistance to the flow of water vapour from the warm side and reduce the flow by incorporating within the insulation a suitable barrier resistant to vapour flow. A second method is by providing suitable openings through which vapour can escape.

It has been established by Moller and Babitt that this vapour barrier should not be permeable to more than about 0.7 grain of moisture/sq.ft/hr/in. of Hg (permeance of 0.7 per ms) under certain conditions.

In the United States and Canada a limit of 1 perm and 0.75 perm respectively for vapour barrier material has been widely used even under severe conditions. In our country a limit of 1 perm has been suggested for various vapour barriers used :

These principles of control of migration of water vapour through refrigeration insulation recommend the following precautions :

(a) Apply the best possible water vapour barrier or seal on the warm side of insulation. Foamglas, Rubatex and Metal-foils or sheets are examples of this type of vapour barrier. Foamglas is usually available – in blocks 12 in.by 18 in.; Ferro-Therm in sheets 24 in. by 32 in; Rubatex in sheets 3 ft. by 4 ft. and Alum : seal in coils 32 in. wide and many feet long. A

Lead-foil adhesive tape is used for making the joints essentially vapour-tight. Asphalt is also used to seal the warm side.

(b) Do not place a vapour seal on the cold side of refrigeration walls if the room is to be continuously maintained at a low temperature.

(c) Since it is nearly impossible to have a perfect vapour seal, good practice is to choose an insulation that is least affected by moisture penetration, provided that the other properties are suitable.

Control of flow of Water Vapour through Insulation under Special Refrigeration Construction :

(a) *Cold Storage Construction :* The vapour barrier is placed on the outside and moisture accumulates during long period of continuous operation. Two parts of the insulation should be joined with bitumen adhesive.

(b) *Reversed Flow Conditions :* Cold Storages operating at temperature aboun 5°C experience such reversals in vapour flow from winter to summer and causes an awkward problem. In special cases vents are provided on both sides of the wall and alternatively opened and closed from season to season.

(b) *Insulated Roofs :* The most important moisture problem encountered it the insulated roofs is of impermeable coverings laid on flat boards or slabs. It is seldom possible to prevent breathing of space occupied by the insulation. This action can be prevented by providing vents to the outside.

Conclusions

Water vapour diffusion through a layer of insulating material due to both temperature and pressure gradients, causing well known serious problem, has led to study of control of flow of water vapour. This control can be achieved in many applications of insulation without much difficulty. Such control is essential for the proper functioning of refrigeration insulation and also to avoid deterioration of environments. Moisture migration control through insulations for many situations have not been fully studied and thus require intensive study to find suitable barriers and their applications.

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Cryogenic Insulation Techniques

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The paper discusses the various types of cryogenic insulations, their relative performance and the fabrication of devices using these insulations.

Superinsulation

The simplest insulation commonly used is just an evacuated medium. Radiative heat transfer causes large evaporation losses. Introduction of a certain number of reflecting layers or radiation shields has improved the performance of evacuated dewars by several orders of magnitude. These radiation shields are separated by means of spacers such as low density paper or netting (0.6-6 mil thick) by crinking or embossing the shields.

Reflecting Shell Assembly

An elegant alternative to such a reflecting layer insulation is that of reflecting packed hollow glass spheres. These spheres are about 15–150μ in diameter and coated on the exterior with aluminium.

Multishielding Techniques

Another novel technique developed in recent times is that of multishielding. The basic idea is to use the cold gas resulting from solid sublimation to intercept and extract a significant portion of the heat leak through the insulation before it reaches the solid cryogen. Highly conductive metallic conductor shields are connected directly to the neck tube through which the cold gas is vented.

Efficiency of Insulation

The efficiency of a cryogenic insulation is the factor S/KD, where S determines the insulation simplicity, ruggedness to transport and handlings, K thermal conductivity and D the packing density of the insulation.

Foam Insulation

Expanded polystyrene beads have got low thermal conductivity due to entrapped air. Hence they are good insulation for liquid nitrogen use. Expanded polystyrene vessels which are more robust to handle and have fairly low thermal conductivity are commonly employed to store cryogens.

Evacuated Powder Insulation

Another type of insulation used is the vacuum powder and glass wool

insulation. It was found that by filling the space with powder, conductivity would be decreased with the initial lowering of pressure. The powders are of such small size that inter particle contact may be considered as point contact. A unique problem associated with powder insulation is compaction of powder in the insulation space.

In the fabrication of any of these insulants, the bonding materials between the several layers also play an important part in determining the overall thermal conductivity.

	Insulating material	*Thermal conductivity*
1.	Evacuated Dewars	10^{-5} mw cm^{-1} K^{-1}
2.	Hollow Glass Spheres	$(2-4) \times 10^{-4}$ mw cm^{-1} K^{-1}
3.	Multi Shields	—
4.	Foam Insulation	1.4 times air thermal conductivity
5.	Evacuated Powder Insulation	$(5-10) \times 10^{-4}$ mw cm^{-1} K^{-1}

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SESSION IV

Machinery and Equipment used in Cold and Freezer Storage

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An Ultra-Freeze Chamber Using Liquid Nitrogen

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Freezing as a means of preserving food has got a long history. The difficult aspect of this technique was to keep the original freshness, taste, texture, flavour etc. of the item during the freezing operation.

It has been proved that the rate of removal of heat from the food products has considerable effect on their physical and chemical properties, which are the criteria for the ultimate quality.

Cryofreezing can provide the fastest freezing rate and in many a case a superior product, almost on par with the original quality of the food.

To appreciate the advantages of using liquid nitrogen as a tool for freezing operation, a basic knowledge of its properties is needed. Nitrogen, a gas at the normal temperature and pressure, is inert to all food substances and has a boiling point of —320°F. From the heat transfer equation $Q/A = U\Delta t$ where U is the heat transfer coefficient and Δt the temperature gradient between the product and the refrigerating medium, it can be seen that, even with a small heat transfer coefficient, liquid nitrogen is capable of effecting a high heat transfer rate. The liquid, when it evaporates, takes 86 B.T.U./lb as latent heat. The nitrogen gas so formed at a temperature of —320°F has got a sensible heat of 0.25 B.T.U./lb per degree temperature rise. It means that if one pound of liquid nitrogen is raised to a temperature of 40°F, its refrigerating power will be $86 + 0.25\,(320 + 40) =$ 176 B.T.U./lb. In addition, the large liquid to gas volume ratio of about 600 provides an effective cooling medium. Very little energy is required for fans to enhance convective heat transfer in this case. For one ton of refrigeration, 0.15 h.p. only is required as convective horse power in the cryogenic freezing compared to 1 h.p./ton of refrigeration in the case of conventional refrigeration.

There are several cases where cryofreezing stands out :

1. When the product itself is very costly and quickly perishable, for example, shrimp. In the conventional system, there is a loss of 4 to 5% weight due to dehydration, whereas in cryofreezing it is only 0.25%.

2. If the product has high water content, cryofreezing gives a better quality food as in the case of tomatoes, melon slices, berries, citrus fruits, etc.

3. In products like bakery items staling takes place in the temperature range of 60°F to 20°F. To inhibit this action, it is necessary to have a high freezing rate.

Cryofreezing with liquid nitrogen takes about 80 seconds, whereas conventional freezing takes more than 120 minutes.

4. The food freezing system using liquid nitrogen will provide an inert atmosphere for the foodstuffs. The packing of food in this atmosphere will again ensure its good quality.

In a cryogenic freezing system there are four phases.

(*a*) *Pre-cooling*

The product is reduced from ambient temperature to some optimum temperature just before entering the freezer. This increases the efficiency of the freezer system and also prevents the food from thermal shocks which would lead to cracking.

(*b*) *Pre-chilling*

The temperature of the product is reduced to the point where latent heat of fusion of the moisture content is removed.

(*c*) *Freezing*

The temperature is brought down below 0°F at two thirds the distance from the outer shell, to a temperature of apparent freezing point for the product. This is accompanied by super cooling in the outer shell, depending on the thermal properties of the product being frozen.

(*d*) *Equilibrium*

The temperature is stabilised at 0°F.

The quantity of liquid nitrogen used in a cryofreezing system depends on product property and input temperature of product, external heat leak into the freezer, freezer efficiency, and nitrogen storage and transfer losses.

(*a*) *Product load*

For most of the food items the consumption pattern of liquid nitrogen is of the order of 1.0 to 1.5 lb of liquid nitrogen/lb of food. As an illustrative example we calculate the product load for shrimp[1]. The temperature is taken to be 40°F and the water content as 83%. The freezing point is 28°F. The specific heat above the freezing point is 0.86 and below the freezing point is 0.45. The product has to be cooled to the equilibrium temperature of 0°F. The quantity of product is taken to be 10 lb/hr (The load due to water glaze is negligible).

Calculation of theoretical refrigeration per pound of shrimp :

To cool shrimp to 28°F	= (0.56) (40.28)
	= 10.3 B.T.U.
To freeze shrimp at 28°F	= (0.83) 144
	= 119.6 B.T.U.

To cool shrimp to 0°F = (0.43) (28.0)
= 12.6 B.T.U.

Total refrigeration required per pound of shrimp

a + b + c = 10.3 + 119.6 + 12.6 = 142.5 B.T.U.

Refrigeration required for 10 pound per hour load,

= 142.5 × 10 = 1425 B.T.U. per hour.

(*b*) *External heat leak into freezer*

To reduce the external heat load and to have a controlled freezer temperature, a good insulation for the freezing chamber is essential. The use of powder insulation evacuated using a mechanical rotary pump is better than vacuum alone, when the insulation thickness exceeds the 'optically thick' limit. Perlite is one of the best insulation powders for this purpose.

To find out the thickness of insulation and the heat leak through it, it is necessary to analyse the three components of this heat leak, namely, (*a*) gaseous conduction through voids between powders, (*b*) solid conduction through contact points in the powder, and (*c*) radiation through the insulation.

In an evacuated powder system, the contribution due to gaseous conduction is negligible. The actual process of solid conduction in evacuated powders is still under investigation and no correct formulae are available. But experimentally it is found to be about 10^{-9} B.T.U. per hour—ft.—°F. For insulation between liquid nitrogen temperature and ambient, radiation is the biggest contributing factor. To minimise the radiation heat input the thickness of insulation should be greater than the optically thick limit. For perlite with an extinction coefficient value of $\beta=256$ inch^{-1}, the thickness should be greater than 0.39 inch[3]. Taking into account the solid conduction and practical experience, a minimum thickness of 2″ is given to the insulation. Since the particle size and density affect the solid conduction and radiation, they are chosen for optimum performance. Perlite having 30 lb/ft^3 density (having 400 micron size) is selected.

The external heat leak through the cold chamber can be calculated as

$$Q = K_a \frac{A}{\Delta x} \Delta t$$

where K_a is the coefficient of heat transfer in B.T.U./hr — ft — °F, A = area of body walls in ft^2, Δx = thickness of insulation in ft, Δt = temperature difference between inside and outside the freezer in °F,

For the freezer described below, the size is 3′ × 3′ × 2. A = 42 ft^2.

Δt at 80°F ambient = [80 — (—320)] = 400°F.

$\Delta x = 2/12 = 1/6$

The value of K_a is 8×10^{-4} B.T.U./hr — ft — °F. Since in this design the optimum density and particle size are chosen, taking K_a from this graph is a safer approach.

Therefore external heat leak Q =

$= 8 \times 10^{-4} \times 42 \times 6 \times 400$

$= 80.64$ B.T.U./hr

Total of product load and external heat leak

$$=1425+80.64$$
$$=1505.64 \text{ B.T.U./hr.}$$

Assuming the overall efficiency of the freezer system to be 80% to account for liquid nitrogen storage loss and transfer loss, exfiltration of nitrogen gas from the freezer, infiltration of air from outside etc.

The total load $=1505.64\times 100/80$
$=1880$ B.T.U./hr.

Quantity of liquid nitrogen required per hour
$=1880/176$
$=10.6/$lb per hour.

The freezer construction and working details

To attain maximum efficiency and to avoid cracking of foodstuffs due to thermal shock, it is necessary to have a pre-cooling chamber before the freezing chamber, which uses the exhaust cold gas from the freezer. The freezing chamber has dimension of $3'\times 3'\times 2'$ and the pre-cooling chamber $1'\times 3'\times 2'$. These are made of stainless steel AISI 304 of 1/16″ thickness, seam-welded using argon arc. As described above the insulation is made of evacuated perlite 2 inch in thickness. A vacuum valve with filter is attached to one side for evacuation of the insulation. The food items are kept on perforated trays in the freezing chamber and pre-cooling chamber. Side doors and interlinking doors are provided to introduce and remove foodstuff.

Liquid nitrogen is stored above the freezing chamber in a commercial 25-litre liquid nitrogen container. The dewar is pressurised using the evaporating gas in the freezing chamber. When the pressure in the freezing chamber exceeds 6 p.s.i it escapes through a series of pop-off valves operating into the pre-cooling chamber. Since small exfiltration into the pre-cooler is already accounted for, the pop-off valve is disc sealed acting on spring stiffness.

The temperature in the freezing chamber can be monitored and controlled using a temperature controller which operates a valve to communicate the freezer pressure to the pneumatic valve in the discharge line from the liquid nitrogen vessel. The refrigerant is discharged through a 3/8″ I.D. copper tube branching into two spray headers at the top and bottom of the freezer. The spray header is in the form of a copper cap with 1/16″ diameter holes drilled at intervals of 6″ which acts as a fine nozzle. The holes are drilled at an angle of 45° to the tube, so that from the top header liquid nitrogen is sprayed down and from the bottom header it is sprayed up.

When the temperature in the freezer is above the set value, the temperature sensor activates the valve in the temperature controller. Thus the freezer pressure is communicated to the pneumatic valve in the liquid nitrogen discharge line and it allows nitrogen to be sprayed into the chamber. The spray of liquid evaporates when it comes into contact with the warm gas. Turbulence is created due to the sudden expansion in volume. To further enhance the heat transfer, a 1/16 h.p. fan is mounted in the freezer, the motor being fixed outside the chamber. To speed up the response to cut off and resumption of liquid nitrogen, by the temperature sensor

action, a U-bend is incorporated in the discharge line. Once the temperature is set, the rest of the operations are automatic and do not require any attention. The design of the temperature controller discussed in many references has been modified to suit this requirement.

A manual control shut-off valve is provided in the discharge line to stop the supply of liquid nitrogen when the doors are opened. The safety valve in the liquid nitrogen supply line ensures safe operation. The complete unit is mounted on a trolley to make it portable.

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Evaporator Hunting in Vapour Compression Refrigeration System

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This paper reports a study of the problem of evaporator hunting in vapour compression refrigeration system. The phenomenon of hunting has been described in terms of the superheat signal Conditions for hunting and non-hunting systems have been found out experimentally, for a horizontal copper tube test evaporator, 6.35 mm I.D., 4.27 m long, using Refrigerant —12. Experimental results indicate that the evaporator hunting is marked by surface temperature fluctuations, causing superheat signal to become unstable. A minimum stable superheat signal ensures a non-hunting evaporator. The minimum stable superheat is found to increase with the increase of evaporator load.

The increase of evaporator pressure is found to increase the minimum stable superheat signal. The effect of inlet vapour quality has been observed to be significant only up to a value of 0.3.

Introduction

It is well known that the overall performance of a vapour compression refrigeration system depends to a great extent on the performance of the evaporator. The full utilization of the capacity of direct expansion type of evaporators has long been a problem with the refrigeration industry. The reason is the instability or hunting in a feed back control loop, consisting of a thermostatic expansion valve and evaporator. Such a control loop is shown in Fig. 1. The instability occurs mainly due to varying quantity of liquid supply through the thermostatic expansion valve to the evaporator. The superheat signal from the thermal bulb adjusts the opening of the valve either to increase or decrease the flow rate of liquid refrigerant. Depending on the evaporator load and evaporator pressure, the refrigerant is evaporated

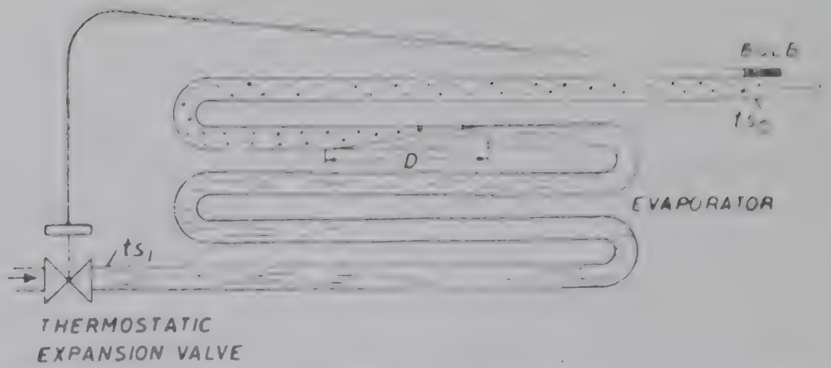

Fig. 1. Feed back control loop of evaporator and thermostatic expansion valve

either partially or completely. The surface temperature at the exit of evaporator, where the thermal bulb is located, is responsible for sending the superheat signal. Depending on this signal the evaporator shall be either overfed or underfed.

The possibility of compressor damage due to liquid refrigerant entering in it in the case of overfeeding and incomplete utilization of evaporator heat transfer surface area in the case of underfeeding are both undesirable from the point of view of maintenance and operation. Unless the valve and evaporator characteristics match each other, the overfeeding or underfeeding of the evaporator shall continue thereby causing instability or hunting of evaporator.

The aim of this paper is to study the phenomenon of hunting and find out conditions of non-hunting system. For this purpose the minimum stable superheat signal approach as proposed by Huelle[1] has been adopted. Though, both the evaporator and expansion valve characteristics are responsible for hunting, only evaporator has been chosen for this study.

Concept of Superheat Signal

The difference between the evaporator tube wall temperature and the saturation temperature at evaporator exit pressure is termed the superheat signal. It plays an important role in the stable operation of the evaporator and determines the utilization of the evaporator heat transfer surface. Fig. 2 schematically shows the variation of refrigerant, of evaporator surface and of the cooling medium along the length of the evaporator. Superheat signal has been shown as Δ t. The surface temperature starts increasing from the point D_1 and becomes steady after D_2. The distance from D_1 to D_2 is the transition zone corresponding to the portion D in Fig. 1. Wedekind and Stocker (2), in their study of the transition zone, found that the surface temperature in this zone fluctuates rapidly and the positions of the transition zone and the final evaporating point D_2 depend on the adjustment of the liquid supply to the evaporator.

With the increased amount of liquid entering the evaporator, the transition zone shifts towards the evaporator exit thereby decreasing the superheat signal. Similarly, by reducing the liquid supply, the transition zone shifts away from the evaporator exit and the superheat increases. Thus the superheat signal is inversely proportional to the liquid supply.

A critical condition arises when the liquid supply is such that the thermal bulb

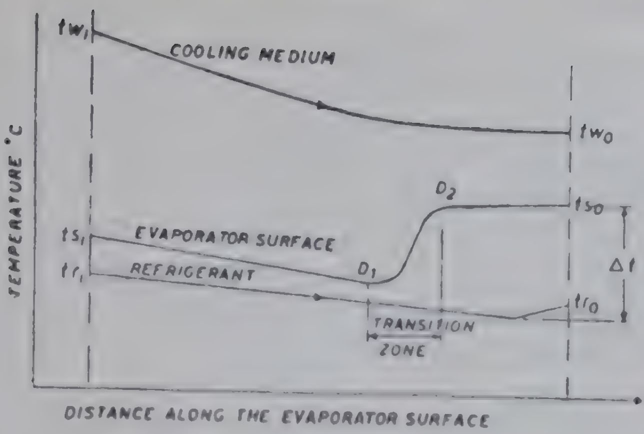

Fig. 2. Temperature variation along evaporator length

of the thermostatic expansion valve is in the transition zone. The surface temperature becomes unstable and results in a constantly varying refrigerating effect. The superheat signal at this stage becomes unsuitable. In order to ensure a stable operation of the evaporator with maximum yield, a minimum stable superheat signal is necessary. Evaporator and thermostatic expansion valve characteristics may be matched to achieve a non-hunting system by the use of minimum stable superheat signal.

Experimental Apparatus

The experimental apparatus as shown in Fig. 3 consists of a complete vapour

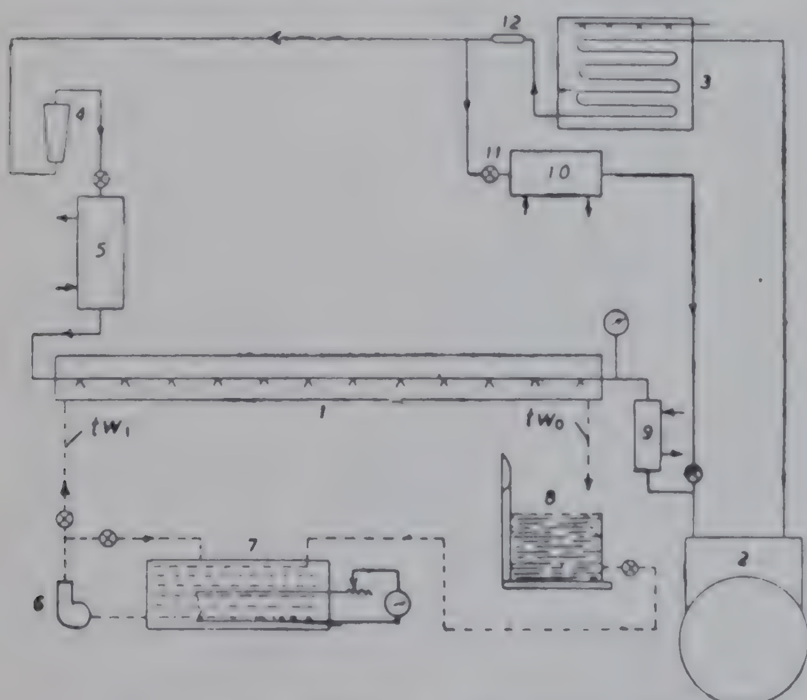

Fig. 3. Line Diagram of Experimental Set-up

1. Test Evaporator	5. Pre-Heater	9. After Heater
2. Compressor	6. Water Pump	10. Auxiliary Evaporator
3. Evaporative Condenser	7. Water Heating Arrangement	11. Throttle Valve
4. Variable Area Flow Meter	8. Water Weighing Arrangement	12. Drier

compression refrigeration system. The test evaporator consists of a 4.27 m (14 ft) long horizontal copper tube, 6.35 mm (¼ in) I.D., through which Rrfrigerant-12 flows and evaporates. The evaporator tube is surrounded by a 25.4 mm (1 in) I.D. copper pipe, through which the hot water flows. Copper constanton thermocouples mounted on the evaporator tube surface at an equal distance of 15.25 cm apart give the temperature of the surface of the evaporator along its length. Deep well thermometer pockets have been provided at the inlet and outlet of the pipe carrying water to the test evaporator. A rotameter measures the flow rate of refrigerant entering the test evaporator. For controlling the quality of the refrigerant mixture entering the test evaporator, a pre-heater has been provided. An after heater at the end of test evaporator ensures the superheated state of the refrigerant entering the compressor. The compressor is vertical, two cylinder, 63.5 mm bore, 76.2 mm stroke and runs at 455 rpm, by a 5 h.p. electric motor. An evaporative condenser fitted with induced draft fan and centrifugal water pump has been used. An auxiliary evaporator along with a throttle valve is used for controlling the flow rate of refrigerant through the test evaporator. The test evaporator has been insulated with 40 mm thick thermocole on all sides to prevent the heat losses.

Test Procedure

The tests were performed under the steady state condition of the system. Flow rate of both the refrigerant and the heated water were maintained constant for a particular run. Also the temperature of the hot water entering the test evaporator was held constant to within 0.2°C with the help of a variable transformer and thermostat provided for this purpose.

For different flowrates of the refrigerant, the evaporator pressure was maintained constant by adjusting flow through the auxiliary evaporator.

The following data were recorded during a test run :

1. Evaporator tube surface temperatures along the test length.
2. Liquid refrigerant flowrate.
3. Pressure at the exit of evaporator.
4. Water temperature entering and leaving the test evaporator.
5. Flowrate of water through the test evaporator.
6. Temperatures of refrigerant at the inlet and outlet of pre-heater and at the inlet and outlet of evaporator.
7. Temperatures of water entering and leaving the pre-heater.
8. Flowrate of water through the pre-heater.

In each test run three sets of data were recorded at intervals of five minutes. All calculations were then based on values obtained by averaging the data from these three sets. Evaporator hunting was marked by the fluctuations of the evaporator tube surface temperature at exit.

These fluctuations were recorded by means of a Digital Data Logging Systems, at the interval of every two seconds.

Results and Discussion

Fig. 4 shows an instantaneous record of the evaporator tube surface

temperature at exit during hunting. From the record it is evident that the temperature fluctuates in a random manner. The superheat signal also, therefore, changes accordingly. The maximum and minimum variation of superheat under the unstable conditions is also shown in the same figure. The exit evaporator surface temperature becomes steady when hunting is absent. In this case superheat has a constant value which is greater than the maximum value of unstable superheat.

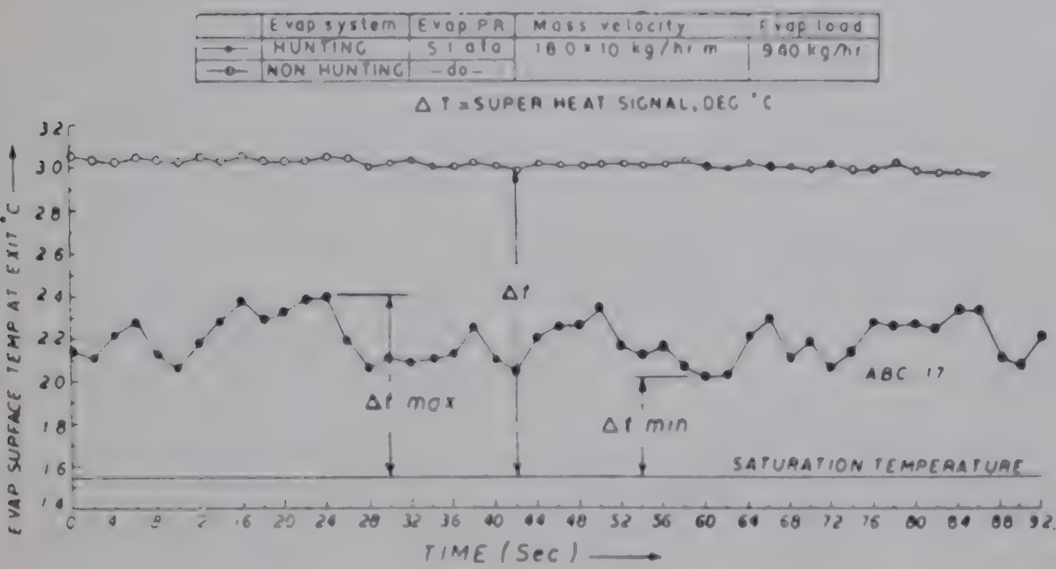

Fig. 4. Evaporator Surface Temperature

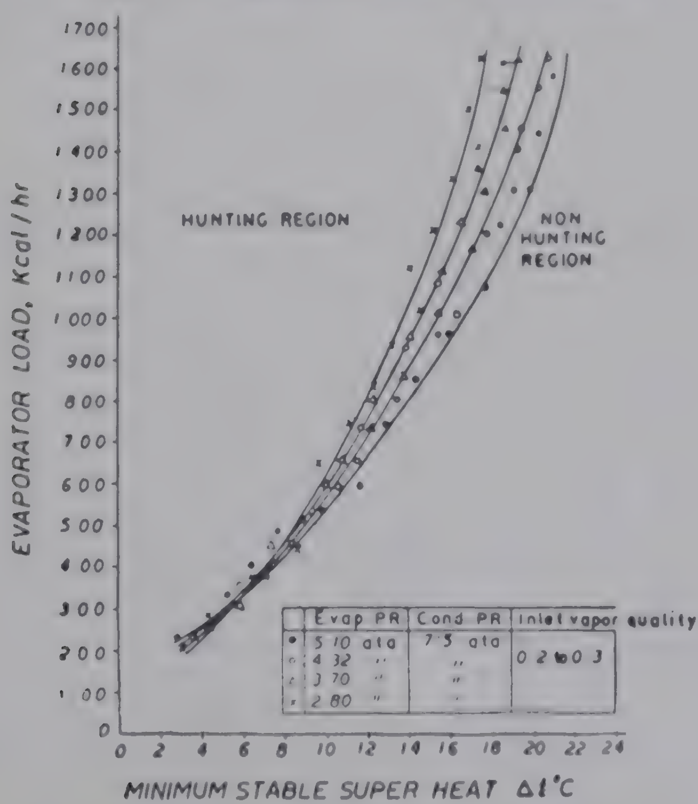

Fig. 5. Minimum Stable Superheat Vs Evaporator Load

Fig. 5 shows the minimum stable superheat plotted as abscissa against the evaporator load as ordinate. The minimum stable superheat has been taken to be that value of superheat signal below which it becomes unstable. The condenser pressure and inlet vapour quality have been kept constant. As seen from the figure, the minimum stable superheat increases with the increase of evaporator load.

For smaller evaporator loads the effect of change of evaporator pressure is non-existant, whereas for evaporator loads greater than 500 kcal/hr, it is notable. It increases the minimum stable superheat. The increase of minimum stable superheat means that the effective utilization of the heat transfer surface area of the evaporator decreases. As soon as the minimum stable superheat is less than that shown by the curve for a particular pressure, the evaporator starts hunting. However, a non-hunting system is possible if the superheat signal is greater than the minimum stable superheat.

All the curves in Fig. 5 have been plotted for approximately constant inlet vapour quality. The effect of change of inlet vapour quality on the minimum stable superheat is shown in Fig. 6 in which the evaporator and condenser pressures are constant and evaporator load is one of the changing parameters. The inlet vapour quality varies from 0.16 to 0.54.

As observed from Fig. 6, the minimum stable superheat shows a tendency to decrease upto a vapour quality of 0.3 and thereafter remains approximately constant. This behaviour is expected due to the decrease of amplitude of fluctuations of the transition zone itself. It has also been observed by Wedekind and Stokers[2] that average amplitude of oscillations of the final evaporating point decreases as the inlet flow quality increases.

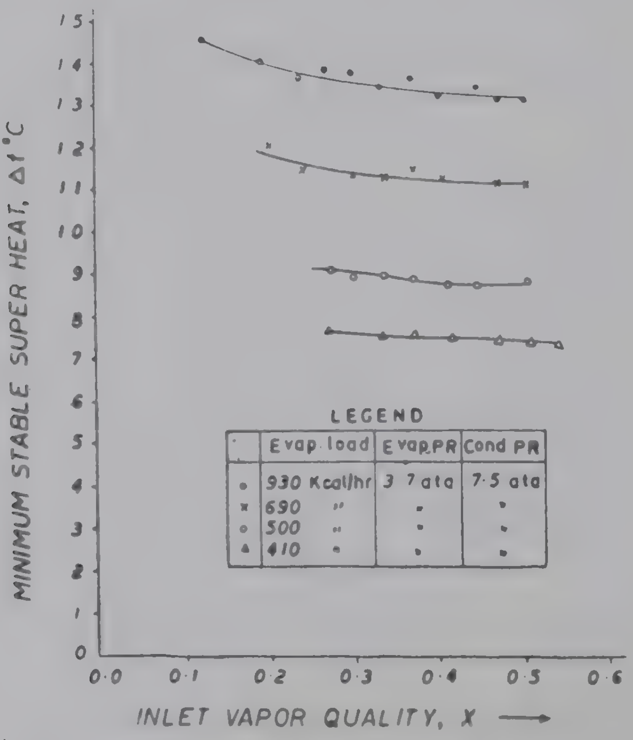

Fig. 6. Effect of Inlet Vapour Quality

Conclusions

From the experimental results stated above the following conclusions can be drawn :

1. The evaporator hunting is marked by the surface temperature fluctuations. At this time the superheat signal also becomes unstable. Any further increase of the liquid supply to the evaporator will soon reduce the value of superheat to zero and evaporator will be overfed.
2. The minimum stable superheat increases with the increase of evaporator load. It also increases with the evaporator pressure at a given evaporator load.
3. The effect of inlet vapour quality on the minimum stable superheat is significant only at lower values of vapour quality.
4. Since a lower minimum stable superheat is desirable, it is recommended that the evaporator should be fed with lower inlet quality. However, it is not always possible. In most of the cases the inlet quality is governed by the flowrate of the refrigerant and the evaporator pressure.
5. In commercial refrigerating systems, where the evaporator liquid supply is adjusted through a thermostatic expansion valve, the system stability will depend on both the evaporator and valve characteristics. Further investigation is needed to study the combined characteristics of the evaporator and thermostatic expansion valve feed back loop.

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Computer Simulation of DX-type liquid Chillers

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Introduction

Though DX-Chillers are being used in the refrigeration industry since long, no attempt has been made to improve the method of predicting the performance of such chillers. A need was therefore felt, of having a method, preferably computer oriented, for carrying out the simulation of DX-Chillers so that influence of all the pertinent factors like variable heat transfer coefficient on refrigerant side, the variation of mass flux in different passes, and fall in refrigerant pressure (and temperature) during the flow, is automatically incorporated. The present paper gives the details of the method which has been developed to fulfil all these requirements.

The Mathematical Model

A typical industrial DX-type liquid chiller is essentially a multipass shell and tube heat exchanger with refrigerant evaporating inside the tubes and taking away the heat from the surrounding coolant. A general method of predicting the dynamic characteristics of such a chiller has previously been presented by the authors[1]. Since this method did not take into account the effect of pressure drop of refrigerant during evaporation it has been modified. The steady state analysis based on this modified method, considering first the case of single pass parallel flow evaporator, is presented. The equations for the general multipass case will then be obtained from the equations for single pass chiller.

Fig. 1 shows the schematic diagram of a simple 1-pass parallel flow evaporator. Considering an infinitesimal length δ_y of the chiller, we write down the various energy mass and momentum balances. The energy balance yields followed by differential equation for the evaporating fluid can be written.

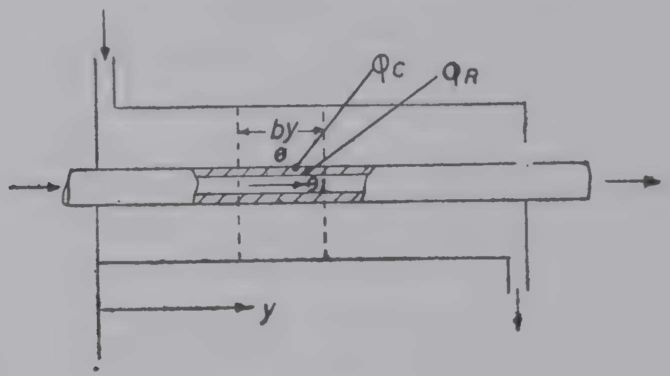

Fig. 1.

$$\frac{V_1 A_1 h_1}{\bar{v}_1^{\,2}} \frac{d\bar{v}_1}{dy} + P_1\alpha_1(\phi_{R_1} - \theta_1) = \frac{V_1 A_1 \dfrac{dh_1}{dy} + A_1 h_1 \dfrac{dv_1}{dy}}{v_1} \tag{1}$$

The continuity equation for the evaporating refrigerant can be written as

$$v_1 \frac{d\bar{v}_1}{dy} = \bar{v}_1 \frac{dv_1}{dy} \tag{2}$$

Using equation (2) in the equation (1), we can further simplify the latter, to obtain the energy equation

$$P_1\alpha_1(\phi_{R1} - \theta_1 = \frac{(v_1 A_1)}{(\bar{v}_1)} \frac{dh_1}{dy} \tag{3}$$

Similarly, the energy balance for the shell fluid element can be written in the differential form as follows:

$$\frac{do}{dy} = \frac{P\alpha\,(\phi_{c_1} - \theta)}{A\,v\rho\,C} \tag{4}$$

The energy balance for the tube can similarly be written and solved for temperature drop across the tube walls. The equation thus obtained is

$$\phi_{c1} - \phi_{R1} = \frac{\theta - \theta_1}{\dfrac{k}{1.0 + \left[ln \dfrac{r_{21}}{r_{11}}\right] \alpha r_{21} \left[1.0 + \dfrac{r_{21}}{r_{11}} \dfrac{\alpha}{\alpha_1}\right]}} \tag{5}$$

The energy balance equation for the tube walls can also be written finally as

$$(\theta - \phi_{c1})\, \alpha\, r_{21} + (\theta_1 - \phi_{R1})\, \alpha_1 \cdot r_{11} = 0 \tag{6}$$

Equation (2) — (6) thus form the basic mass and energy conservation equations for the chiller. For the sake of generality and ease of solution, these equations have been non-dimensionalised by using suitable reference variables as mentioned in nomenclature. The new variables are now defined as

$$1 = \frac{y}{H},\ \tau = \frac{t}{H/v}\ \ v_1' = \frac{v_1}{v},\ \theta' = \frac{\theta}{\theta_o}\ \ \theta'_1 = \frac{\theta_1}{\theta_o}$$

$$\phi c'l = \frac{\phi\, cl}{\theta_o},\ \phi\, Rl' = \frac{\phi\, Rl}{\theta_o},\ \alpha_1' = \frac{\alpha_1}{\alpha},\ X_1 = \frac{\bar{v}_1}{\bar{v}_L},\ H_1 = \frac{h_1}{\lambda} \tag{7}$$

and the resulting dimensionless equations are

$$v_1' \frac{dX_1}{d_1} = X_1 \frac{dv_1'}{d_1} \tag{8}$$

$$v_1' \frac{dH}{d_1} = a_{11}\, X_1\, \alpha_1' (\phi R_1 - \theta_1') \tag{9}$$

$$\frac{d\theta'}{d_1} = a_{31} (\phi c_1 - \theta') \tag{10}$$

$$\phi c_1' - \phi_1' = \frac{\theta' - \theta_1'}{\dfrac{k}{1.0 + ln\left[\dfrac{r_{21}}{r_{11}}\right] \alpha\ \ r_{11}\left[\ 1.0 + \dfrac{r_{21}}{r_{11}\alpha'}\right]}} \tag{11}$$

$$(\theta' - \phi c_1') + (\theta_1' - \phi' R_1)\, \alpha_1' (r_{11} / r_{21}) = 0 \tag{12}$$

where the a's are dimensionless constants defined as

$$a_{11} = \frac{\alpha\, P_1\, \theta_o\, v_L\, H}{v\ A_1\ \lambda} \qquad a_{31} = \frac{P\, \alpha\, H}{A\ v \rho c} \tag{13}$$

Till now momentum balances being advertently avoided to get expressions for the pressure drop of either fluid. However, it is almost impossible to write down the momentum balances explicitly since they are influenced by the highly intricate factors like configuration of baffles, the various clearances between baffles and shell and between tubes and tube holes, the physical nature of the tubes, and the complex interactions between the liquid and vapour phases of the evaporating refrigerant. Therefore the experimentally verified, semiempirical correlations have been used to obtain the pressure gradient of both the coolant as well as the evaporating refrigerant. Thus, for the refrigerant we can write,

$$\frac{dp_1'}{d_1} = f(H_1, v_1', \theta', \phi_{R1}') \tag{14}$$

where p_1', is the non-dimensional pressure of the refrigerant, given as,

$p_1' = p_1/p_o$

Similar expression can be written for the coolant pressure gradient, but it is not necessary since this pressure drop does not directly affect the thermodynamic performance of the chiller.

Equations (8)—(14) thus form the fundamental equations governing the steady state performance of this one-pass chiller. The extension of these equations to multipass chillers is quite straightforward. Thus, considering a two-pass chiller, shown schematically in Fig. 2, we have in the first refrigerant pass, the flow of refrigerant parallel to the shell flow so that its energy conservation equation is exactly the same as equation (9). However, the flow in the second pass is in —y direction, with reference to our y-axis. Hence when we write the energy balance for the evaporating liquid, as done before, the sign of the convective energy transport term is reversed, so that the final equation is

$$-v_2' \frac{dh_2}{d_1} = a_{12} x_2 \alpha_2' (\phi_{c2}' - \theta_2') \tag{15}$$

the continuity equation, however, becomes

$$v_2' \frac{dx_2}{d_1} = x_2 \frac{dv_2'}{d_1} \tag{16}$$

In the shell fluid heat balance we now have the coolant losing its heat to the refrigerant in both the phasses, so that its energy balance gives

$$\frac{d\theta'}{d_1} = a_{21} (\phi_{c1}' - \theta') + a_{21} (\phi_{c2}' - \theta_2') \tag{17}$$

The steady state energy conservation equations for the tube walls of the first pass are obviously the same as equations (11) and (12), and those for the tubes in second pass can be similarly written as

$$(\theta' - \phi c_2') + (\theta_2' - \phi R_2') \alpha_2' (r_{12}/_{22}) = 0$$

$$\phi c_2 - \phi R_2' = \frac{\theta' - \theta_2'}{\dfrac{k}{1.0 + ln\left[\dfrac{r_{22}}{r_{12}}\right] \alpha\, r_{22} \left[1.0 + \dfrac{r_{22} / r_{12}}{\alpha_2'}\right]}}$$

The pressure gradient for the refrigerant flowing through the tubes in the second pass of the chiller can also be found out from the relationship given by equation (14), so that we can write :

$$\frac{dp_2'}{d_1} = f\,(H_2, v_2', \phi R_2) \tag{19}$$

Equations (8), (9) and (11) — (19) are thus the fundamental equations governing the steady state performance of the 2-pass chiller of Fig. 2.

Proceeding in a similar manner, we can write down the system of equations governing the performance of a n-pass chiller

$$v_i' \frac{dH_i}{d_1} = \pm a_{1i} X_i \alpha_i' (\phi_{Ri} - \theta_i') \tag{20}$$

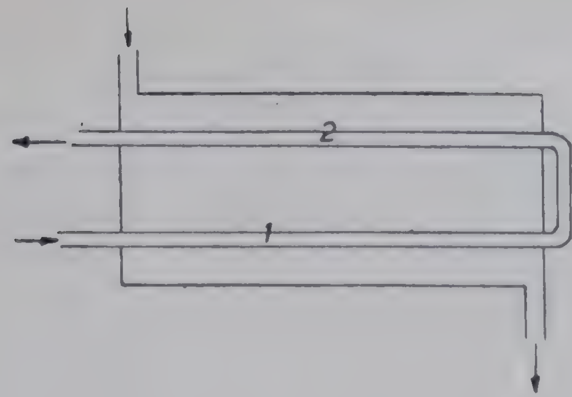

Fig. 2.

$$\frac{d\theta'}{d_j} = \sum_{i=l}^{n\ \mathrm{pass}} a_{3i}\,(\phi'_{ci} - \theta') \tag{21}$$

$$v_i' \frac{dX_i}{d_i} = X_i \frac{dv_i'}{d_i} \tag{22}$$

$$0 = (\theta' - \phi'_{ci}) + (\theta_i' - \phi_{Ri})\,\alpha_i'\,(r_i / r_{2i}) \tag{23}$$

$$\phi_{ci}' - \phi_{Ri}' = \frac{\dfrac{\theta' - \theta_i'}{k}}{1.0 + \mathrm{In}\left[\dfrac{r_{2i}}{r_{1i}}\right] \alpha\, r_{2i} \left[1.0 + \dfrac{r_{2i}}{r_{1i}\,\alpha_i'}\right]} \tag{24}$$

$$\frac{dp_i}{d_1} = f\,(H_i, V_i', \theta', \phi_{Ri}') \tag{25}$$

all for $i = 1, 2, 3 \ldots\ N\,PASS$

Where +ve sign is taken in equation (20) if the refrigerant is flowing parallel to the fluid

Out of these equations, equation (22) can be directly integrated to give

$$X_i = C_i\, v_i' \tag{26}$$

where the constants C_i can be obtained from the known mass flow rate of refrigerant M, since

$$C_i = \frac{X_i}{v_i'} = \frac{\bar{v}_i}{\bar{v}_L}\left(\frac{v}{v_i}\right) = \frac{v\sqrt{\bar{v}_L}}{v_i\sqrt{\bar{v}_i}} = \frac{v\sqrt{\bar{v}_L}}{M/A_i} \tag{27}$$

Thus, equations (20)—(21) and (23)—(26) form the complete system of equations describing a n-pass DX—Chiller.

It can be seen from above that in order to predict the performance of a n–pass chiller, $2n+1$ differential equations and $2n$ algebraic equations have to be solved simultaneously. These equations involve in all $6n+1$ variables viz ϕ_{ci}, ϕ'_{Ri}, α'_i, H_i p'_i, θ'_i (all for $i = 1, 2, \ldots\ldots n$) and θ'. Out of these variables, the $\theta'_i s$ are directly related to the corresponding non-dimensional pressures p'_i by the saturation pressure temperature relationship of the refrigerant used. Again, the non-dimensional local heat transfer coefficients α'_i can be directly calculated from the values of the refrigerant flow rate M, enthalpy H_i and temperatures ϕ_{Ri} and θ'_i by using the

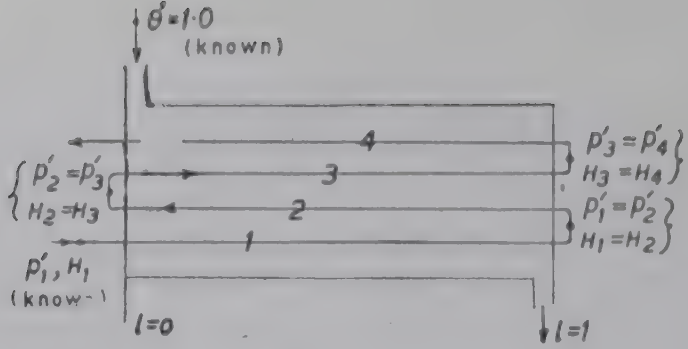

Fig. 3.

experimentally verified, semiempirical correlations developed by various research workers. With only $4n+1$ independent variables left φ_{ci}', φ_{Ri}', p'_i and θ' can be chosen. Again out of these variables it is possible to express the first $2n$ variables as functions of the rest by using the $2n$ algebraic equation (23) and (24). Finally $(2n+1)$ first order differential equations in an equal number of variables (viz H_i', p_i' and θ') are available indicating a complete solution to the problem if $(2n+1)$ boundary conditions to be satisfied by the solution are also made available.

It can be easily seen that for any n-pass chiller, the requisite number of boundary conditions are always available. As a typical example, the boundary conditions for a 4-pass chiller are shown in Fig. 3. Thus, out of the 2n+1 constraints, one corresponds to the known shell fluid entry temperature, another two correspond to the known pressure and enthalpy of the entering refrigerant and the rest are obtained from the conditions of compatibility of the refrigerant pressure and enthalpy at the junctions of the consecutive passes. It can also be seen that at the most 3 of these conditions can be initial conditions, and that it will be so only if the flow of refrigerant in the first refrigerant pass is in the same direction as of the shell fluid. In case it is not so, only one of these conditions, viz, that corresponding to known shell fluid entry temperature, is an initial condition, all the rest being boundary conditions. Thus, in either case we have a non-linear mixed initial and boundary value problem to be solved.

Method of Computation

The solution of these differential equations in a closed form is obviously ruled out. However, there exists numerous numerical methods[2] to solve such problems. The method which we have adopted to solve this system of equations is the 'Initial Value Method for Boundary value problems[2]. In order to have minimum possible computational time while retaining the accuracy, the Adoms Moulton Predictor corrector method[3] have been employed. An iterative scheme based on Warner's modification[4] of Newton-Raphson method has been used to arrive at the correct initial values.

The complete procedure has been programmed for solution on an ICL 1909 digital computer. The program is completely general so that it is possible to predict the steady state performance of any conceivable DX-type refrigeration chiller. In the program suitable equations had to be chosen to represent the dependence of

pressure gradient and heat transfer coefficient of both, the refrigerant side and the shell side, on other variables. After an exhaustive survey of the existing literature, Chawla's[5,7] equations for predicting the heat transfer and pressure drop for evaporating refrigerant have been finally adopted. For shell side pressure drop prediction, the equations of Bell[8] and for shell side heat transfer coefficient recommendations of Whitley[9] and the equations of Kern[10] have been used.

Results and Discussions

In order to check the method developed, the computer program has been used to predict the performance of a typical industrial chiller. The computer was provided with the design data of a 40 ton DX-chiller marketed by a prominent Indian manufacturer and the predicted performance was compared with known data. Thus, a capacity of 40.4 tons is predicted which means an error of about one per cent from the cited value of 40 tons. Similarly, the predicted water side pressure drop is 20.5 ft of water, which is about 15 per cent off from the approximate value of 23.5 ft. of water given by the manufacturer. On the other hand the refrigerant side pressure drop predicted by the simulation procedure is about 2.5 psi as against the approximate value of 2 psi given to us by the manufacturer.

It can be seen that the predicted performance of the chiller agrees quite satisfactorily with its actual performance. The comparatively large error in pressure drop predictions can be attributed to various reasons. Firstly, the pressure drop data as given to us by the manufacturer are certainly not accurate and are prone to errors of almost the same order of magnitude as observed in our predictions. This is so because of the fact that, since the main duty of chillers is to provide sufficient cooling, it is their refrigerating capacity which is generally measured accurately and specified by the manufacturers, and not the water side and refrigerant side pressure drops.

Another factor which could contribute towards these deviations is the difference between the actual values and the assumed values for some of the design data, which were not available. Thus during water side pressure drop calculations it has been assumed that shell to baffle clearance and the tube to baffle clearance areas are negligible, which may not be exactly true. Similarly, for refrigerant side calculations, the relative roughness of the tube surface was taken as 0.001, and this may also not be exactly correct.

Lastly, another factor which is significant in this context is the accuracy of heat transfer and pressure drop equations employed in the program. Though, the best available equations have been used, it is quite natural to expect in such correlations differences between the actual and predicted heat transfer coefficient and pressure gradients, upto about $\pm$ 10 to 15 per cent. Obviously, these errors will also contribute towards the difference between the actual and the predicted performance of the chiller.

Nomenclature

a_{11}, a_{31} : Dimensionless constants defined by equation (13)
A : Total shell side crosssectional area

A_1	:	Total crosssectional area for flow of refrigerant
c	:	specific heat of shell side liquid
c_1	:	Dimensionless constant defined by equation (26)
h_1	:	Refrigerant enthalpy
H	:	Total length of each tube
H_1	:	Non-dimensional refrigerant enthalpy
k	:	Thermal conductivity of tube material
l	:	Non-dimensional length
p	:	Outside perimeter of the tubes
p_1	:	Inside perimeter of the tubes
r_{21}	:	Outside dia. of tubes
r_{11}	:	Inside dia. of tubes
t	:	Time
V	:	Shell fluid velocity
V_1	:	Refrigerant velocity
$\overline{V}_1$, $\overline{V}_L$	:	Specific volume of the refrigerant at a distance y, of refrigerant liquid at entry conditions
X	:	Dimensionless specific volume
Y	:	Distance along tube length

Greek Symbols

α	:	Shell side heat transfer coefficient
α_1	:	Heat transfer coefficient of the evaporating refrigerant
δ	:	Incremental quantities
θ, θ_o	:	Shell fluid temperature at any distance, y, at entry
θ_1	:	Refrigerant vapour temperature
ϕ_{R1}	:	Temperature of tube wall, refrigerant side
ϕ_{o1}	:	Temperature of tube wall, coolant side
ρ	:	Shell fluid density
λ	:	Latent heat of vaporisation
ι	:	Dimensionless time

Subscripts and Superscripts

*	:	Dimensionless variables
i	:	Quantities pertaining to the ith pass
o	:	Reference variables for non-dimensionalisation

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A Proposed Rating System for Food Freezing Techniques

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Introduction

Freezing as means of preservation of food products has acquired the status of widespread acceptability. Quality of the frozen foods is continuously improving, thanks to the simultaneous developments in food freezing techniques. The variety of freezing techniques that is available today for freezing and processing of food makes it problematic to choose the right combination of the food product and the most suitable means of freezing.

It is attempted to tackle the problem of finding the right means of freezing for a particular food product. A number of factors are to be considered before choosing the correct combination of food product and means of freezing and perhaps the most rational approach would be to evolve a rating system.

Characteristics of freezing techniques for rating system include (1) Freezing rate in cm/hr. is an indication of the speed of freezing possibly attainable and the time taken for freezing the food product. Faster freezing will be more desirable and depends on several factors such as (*a*) the influence of surface heat transfer coefficient, (*b*) the packing condition of the food product, (*c*) the thermal properties of the food product, (*d*) the geometry of the food product.

(2) Weight losses: Apart from the direct loss in weight the product will be transformed in texture, form, appearance depending on the freezing technique.

Loss of freezant and freezant recovery: In freezing systems using liquefied gases, a percentage of freezant will be lost. Recovery is possible almost complete in liquefied fluorocarbon freezing, partial recovery in carbon dioxide spray system and no recovery in case of liquid nitrogen freezing. There will be loss of brine in case of brine immersion systems.

Mode of freezing: The mode of freezing varies from individual quick freezing to block or slab freezing or bulk freezing to inpackage freezing. Crust freezing is only possible in some cases while core freezing can take place in certain cases.

Output : The output of the freezer depends on the freezing speed and type. Some freezing means are not suitable for continuous operation in a continuous food processing line. Some due to the speed of operation and some by virtue of the freezer design suitable for in time freezing. Liquefied gases and belt fluidized freezing are more suitable for in time freezing, while air blast (tunnel) plate freezers are suitable for batch operation.

Space requirements : Product size requirements impose certain restrictions on the available space. However, freezing can compensate these in certain cases like plate freezing of meat slabs at the cost of increasing auxiliaries like packing equipment and packaging materials, product cutting, slicing, sizing machines, recovery of freezant equipment ; trolleys, trays, storage tanks, transfer pumps etc.

Power requirements : In terms of horse power this depends on the refrigeration requirements and the temperature of refrigeration including auxiliary refrigeration units for freezant recovery, transfer pumps, circulating fans, spray pumps and conveyor belt motion, power for hydraulic units excluding power requirements for liquefaction of gases like N_2, CO_2 which came under freezant consumption.

Freezer power requirements are the lowest for LN2 systems as they need only circulating fans, transfer pumps, freezer belt to run. Power requirements for fluidized freezing and belt freezing are more than air blast freezing (only fan power) as fluidization needs power for hydraulic operation of plates.

Product size and handling : The design of a freezing unit influences the product size it can handle. While large sized products like carcasses of beef, lamb can be handled only by blast air freezing means, regular shapes like flat, slabs, fillets can be conveniently frozen by plate contact ; any type of small to medium sized food product can be processed by liquefied gases like IN2, LCO2 or LFF. Immersion freezing by brines and glycols is very eonvenient for freezing of poultry in packages and bulk freezing of sardines and for operation in sea fishing.

Power requirements and connected horse power : The temperature and velocity of the freezing medium influence the freezing speed but lower temperatures require greater power inputs. Air blast freezing requires about 35 tons of refrigeration for freezing a ton of food product to about —18°C. Requirements of power in case of liquid nitrogen at —196°C are about 10 times higher than blast air freezing at —40°C.

Packaging requirements : The requirements are widely ranging from simple cartons for preventing surface desiccation and for convenience of regular flat shape (for contact freezing) to protective packaging for contamination or preventing freezant absorption in brine freezing. Packaging also prevents likely product splitting, shattering at cryogenic (—196°C) freezing temperature etc.

Freezing cost : All the factors listed above contribute to the total cost of the freezing process. Quality of the product differs in each case but it is seen the quality differences are marginal in certain products. Hence the costs should be viewed, not from quality considerations alone, but several other contributing factors like investment, system complexity, maintenance and labour, reliability, unit compactness. It is only in case of certain products superior quality processing is

likely to fetch proportionately high returns like marine products frozen by liquid nitrogen freezing process.

Types of freezing techniques[5,6]: Freezing media are air, liquid and metal and all the available freezing techniques can be brought under one of these three versions. It is, however, important to point out that each of the freezing techniques available is characterised by widely differing and sometimes unique features and they should be examined with suitable weightage of each feature for each food product.

Rating of freezing techniques : For each food product depends on its kind and origin, geometry. In order to evaluate the food product freezing and the freezing technique a single entity rating system is needed. The rating system must consider the effects of freezing speed, weight losses, product size and other considerations that are listed under characteristics of freezing techniques. The simplest approach to a rating system will be the most usual linear rating. There are freezing techniques and characteristics and can be rated serially 1 to 5. The objection to these methods is that bad is not punished enough and the spread between the good and bad is not large enough to allow an objective appraisal. In addition the food product and freezing techniques are not rated as a single system (By giving liquid nitrogen by virtue of its high speed of freezing say a rating of 5 and air blast freezing say a rating of 1, similar will be the result by giving air blast freezing 5 for its ability to freeze any size product and 1 to 2 for liquid N2 technique for its ability to handle any medium sized product). A look at the table will prove the defects of a linear rating system. (Table 1)

TABLE 1. *Linear Rating*

	FREEZING TECHNIQUE *							
Criterion	1	2	3	4	5	6	7	Remarks
Rate of freezing	1	2	3	4	5	5	5	
Packaging	2	2	5	0	3	3	3	Maximum
Wt. loss	2	4	4	3	4	5	5	Credit of 5
Product size	5	2	1	4	3	3	3	For Each
Connected H. P.	2	5	4	4	0	3	4	Criterion
Rating Tally out of 25	12	15	17	15	15	19	20	
% Rating	48	60	68	60	60	76	80	

* Freezing Techniques : 1. Air Blast 2. Plate 3. Fluidized Bed Belt 4. Brine/Glycol Immersion 5. Liquid Nitrogen 6. Liquid Carbon dioxide 7. Liquid Freon Freezant System

Exponential rating approach : An exponential rating instead of a linear rating system may combine all the factors that influence the freezing performance in the following formula :

$$\text{FPC} = \text{Freezer Performance Coefft.} = \frac{XS^2+XL^2+XH^2+XC^2+XF^2+XK^2+XO^2}{10}$$

P : Rate of freezing (Speed)

S : Product size

L : Loss in weight

H : Horse power requirements

C : Cost

F : Freezant loss

K : Packaging (K influence), Labour, Output

X : 1 or 2 but will be 2 only for 2 items

Each factor would be rated from 0 to 10.

There is a need for proposing this type formula as the linear rating approach does not punish severely enough a system that is almost good enough in each department.

Let us say that : 90 to 100 = A

80 to 89 = B

70 to 79 = C

60 to 69 = D

and below 60 = F

Evaluating in a linear fashion it was found each item was rated 9, the system would have a total of 90 which is an A ; But using the exponential approach we have

$$\text{FPC} = \frac{6(9)^2+2(9)^2+2(9)^2}{10}$$

$$\text{FPC} = \frac{810}{10} = 81\%$$

FPC = B

In the same way a system that averaged 8 would equal

Linear = 80 = B

Exponential = 64 = D

and a system that averages F

Linear = 70 = C

Exponential = 49 = F

Weightage factor : Two characteristic features serve the purpose of stressing the most important aspects in a given situation. Air blast freezing process of carcasses like beef and lamb offers flexibility to handle any product size and thus merits a double rating.

Similarly air blast freezing merits a double rating for its low cost. On the other hand, freezing of marine products like shrimp by liquid fluorocarbon freezant technique affords individual quick freezing of shrimp which merits it a double rating of 10 with practically no loss in weight offering a better quality product in addition to fetching a better price. So the LFF process gets double rating for freezing speed and weight losses.

A given technique may be good for one food product while not necessarily be so for the other. Fluidized freezing will be most suitable for peas as it can be easily fluidized and frozen IQF with minimum to almost insignificant product losses;

but the same cannot be said of freezing meat slabs or packaged fish fillets where plate freezing has distinct advantage and is the fastest and will work out cheapest due to unidirectional freezing. The above considerations are applied to rate the various freezing techniques.

A case study of how the freezing techniques can differ in their ratings for freeze processing in bulk, a fishery product like sardines, can be seen from the following :

First a decision has to be made about two of the most important considerations for weighting factor rating of 2. The products are frozen in bulk. Individual quick freezing will result in a superior product ; bulk freezing as seen from the table could entail a lot of handling. The best freezing technique will have to minimise this.

(Sardines to be frozen : 2 tons/hr.)

Air blast freezing :

(1) Freezing speed : Tunnel freezing will be done. No quick freezing will be possible. So air blast freezing gets a rating 0.

(2) Labour requirements : Maximum personnel are needed (say atleast 20). Here too, air blast gets 0.

(3) Weight loss will be inevitable in an air blast. Freezing in bulk can reduce this loss. Let the rating be 6.

(4) Power requirements : Tunnel freezing requires about 35 tons of RF per ton of Sardine freezing per hour at a temperature of —35°C. This will be in addition to the fan power requirements. Total power could be about 150 KW. But this will be not higher than other liquefied gases. Let the rating be 7.

(5) Compactness of the freezer : Air blast freezer tunnel could be very compact and economic from total space point of view. A rating of 8 will be appropriate.

(6) Freezing cost : The cost of freezing becomes higher than expected because of the high labour requirements and considerable power, water requirements, though no freezant recovery problems exist. A rating of 5 is given.

(7) Freezant loss : No freezant loss problems exist. A rating of 10 is automatically secured.

(8) Auxiliaries : Requires trolleys etc. (in addition to circulating fans). Air systems require less maintenance from corrosion stand point, unlike, say, brine systems. But mechanical equipment may present some problems. Reliability is on the whole good meriting a rating of 8.

Continuous brine immersion freezing :

Freezing rate : Continuous freezing ensures an almost individually quick frozen product. But brine up-take as in spray can present problems in fast IQF but slower circulation may give cluster freezing. A rating of 7 will be a compromise.

Labour : About 5 persons are needed for controlling freezer operation, loading, for looking after brine, water, electricity services. A rating of 7 is given.

Weight loss : May not be significant, in fact salt up-take may increase the weight, but leaching may be noticed. There may be some contamination if brine is not changed continuously or specific gravity is not maintained. A rating of 5 will weigh the considerations.

Compactness : Circulating pumps, infeed hoppers, outfeed chutes etc. will increase the space. Brine tanks, chiller, heat exchangers consume a further space. Though freezer space should be smaller as liquids can have higher heat transfer coefficients. A rating of 4 will be suitable.

Freezant loss : To replace the brine occasionally for avoiding contamination also during maintenance of brine tanks etc. (about 800 m^3). A rating of 5 will be suitable.

Freezing cost : A faster freezing will bring down the total operating expenses and results in a better utilisation of the installation. A rating of 8 will be suitable.

Power requirements : Refrigeration requirements can be lower as brine temperature of —20 to —25°C will be providing the same fast output as air flow. A rating of 9 will be representative of this process.

Auxiliaries : Auxiliaries may require like agitators, pumps, transfer pumps, tanks.

Reliability : Could be good for freezing but the corrosive nature of brine can affect the performance. Rating could be 5.

Liquefied gas freezing : Like liquid fluorocarbon freezant system.

Freezing rate : IQF, continuous, fast. Given a rating of 10.

Labour : About 1 person could be sufficient most of the time. Given a rating of 9.

Product weight loss : Almost insignificant. Rating could be 9.

Compactness : Quite compact as far as freezer is concerned. Auxiliaries require more space. Transfer pumps, tanks etc. A rating of 7 is given.

Power requirements : At about 30 tons of R at —45°C apart from auxiliary power requirements. A rating of 5 could be given.

Freezing cost : Cost of freezing will be quite competitive as power, labour, maintenance charges are minimum. Only significant cost will be freezant consumption. A rating of 8 may be given.

Freezant loss : Good recovery systems have contained the freezant loss ; loss of about 1 kg for a 100 kg may still be taken. A rating of 7 may be considered suitable.

Auxiliaries : Too many of them are required like transfer pumps, storage tanks, auxiliary refrigeration equipment for recovery of freezant. A rating of 4 may be suitable.

The rating of different systems could be summarised as follows:

TABLE 2. *Bulk Freezing of Sardines*

	W.F.	Air Blast Freezing	BIF	LFF
R: Rate of freezing	2	$2(0)^2 = 0$	$2(7)^2 = 98$	$2(10)^2 = 200$
L: Labour	2	$2(0)^2 = 0$	$2(7)^2 = 98$	$2(9)^2 = 162$
W: Weight loss	1	$1(6)^2 = 36$	$1(5)^2 = 25$	$1(9)^2 = 81$
P: Power requirement	1	$1(7)^2 = 49$	$1(9)^2 = 81$	$1(5)^2 = 25$
S: Space compactness	1	$1(8)^2 = 64$	$1(4)^2 = 16$	$1(7)^2 = 49$
C: Freezing cost	1	$1(5)^2 = 25$	$1(8)^2 = 64$	$1(8)^2 = 64$
F: Freezant loss	1	$1(10)^2 = 100$	$1(5)^2 = 25$	$1(7)^2 = 49$
A: Auxiliaries	1	$1(8)^2 = 64$	$1(5)^2 = 25$	$1(4)^2 = 16$
		338	432	646
Final rating		33.8%	43.2%	64.6%

TABLE 3. *Freezing Teehniques Relative Ratings*

Freezing techniques	Food Products			
	Meat	Poultry	Fish/Crustace	Vegetables including Fruits
Air Blast Tunnel	52	47	36	40
Plate Contact	47	54	55	56
Belt (Fluidized)	41	48	48	76
Immersion (Brine/Glycol)	58	66	49	40
LN_2 Liquid Nitrogen	46	43	65	45
LCO_2 Liquid Carbon dioxide	47	51	51	47
LFF Liquid Fluorocarbon	52	61	74	61

Notes ; For table on relative ratings.

The following weighting factors have been taken for a rating of 2.

Product	Criteria to merit rating of 2 for weighting factor
Meat	Labour and Space requirements
Poultry	Weight Losses and Freezant Consumption
Fish (Crustacea)	Weight Losses and Rate of Freezing
Vegetables inclg. fruits	Weight Losses and Freezant Cost.

Conclusion

1. Such an exercise (as shown in the paper) for evolving ratings will be necessary for complete evaluation of freezing techniques and products, when a decision is to be made for arriving at the correct:

Product—process combination.

2. This exercise will be further useful in commercial freezer development and production programmes to find out how "universal" a single freezing process can be.

3. Chances of personal prejudices and preferences influencing decision can be minimised compared to any linear rating evaluation.

4. Rating system proposed serves as a good checking system. After installation and while in operation the various items can be reassured and compared with the estimations made earlier.

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SESSION V

Airconditioning

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High Capacity Air-conditioned Railway Passenger Coaches

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Air-conditioning of railway coaches was one of the most important steps taken in the direction of increased comfort and regular air-conditioned coach services were introduced by the Indian Railways in the early 50s over 20 years ago. These services have become increasingly popular and as a result the pressure of booking has increased to such an extent that a sizeable increase in air-conditioned coaches has become inescapable.

Needs of Growing Services

The majority of coaches now in service have 14 sleeper berths. In 1956-57 Indian Railways introduced the Air-conditioned Express De luxe service with 400V 3 phase ac power supply which enabled introduction of 71 seater air-conditioned chair cars. Thus enhancing the air-conditioned accommodation substantially. Only a few trains of this type are in operation, the two Rajdhani Express trains between Howrah and Delhi and Bombay and Delhi being the latest additions.

Demand for fully air-conditioned trains, however, is rather limited. The real need today is to add one or two coaches to several mail and express trains which can provide a substantially large air-conditioned accommodation the coaches being self-contained with their own power generation equipment, as in the case of other coaching stock. To meet this special requirement design of the following self-contained types of air-conditioned coaches are now being contemplated :

(i) 48 berths two tier air-conditioned sleeper.

(ii) Composite air-conditioned coach with 10 First class sleeper berths and 34 adjustable reclining chair seats.

(iii) Air-conditioned chair car with 71 adjustable reclining chair seats.

These designs have been developed with a view to enhancing the accommodation for upper class passengers in fewer coaches. Thus enabling extra coaches to be attached for use of other passengers. Tentative layouts of accommodation in these coaches are shown at Annexure I.

Equipment Selection and Design

Selection of suitable equipment for building self-contained coaches of the type mentioned, with limited space available for equipment is indeed a very complex problem. In India hermetically sealed or semi-sealed units have not so far been proved suitable for the rigorous duty in train services. The performance of sealed

units in stationary installations is itself not entirely satisfactory. Their use in services involving shocks and vibrations therefore would be rather risky and expensive.

For reasons of safety use of voltage higher than 110V is not considered desirable on passenger coaching stock. Use of dc supply with batteries to maintain the continuity during halts is also inescapable for self-contained coaches. Development of compact designs of sealed or semi-sealed type has therefore not been possible so far. The conventional open type units are, therefore used for these coaches.

The design of the air circulation and distribution system in the very limited spaee available, where some occupants are located very close to the roof ducts while others are at a comparatively greater distance presents considerable problems. To achieve uniformity in the circulation of cool air, several systems such as roof ducts with ceiling air discharge, floor ducts with underseat distribution, single and twin duct systems and several other variations are possible. The design of the air distribution system is governed by the layout of the coach. The subject of air distribution covers a very wide field and, therefore, need special attention in the design and selection of equipment.

Design Requirements for Air-conditioning

While evolving the design of the coaches, which ply over very long distances of over 2000 km from one end to the other end of the country, the wide variations of temperature and humidity conditions have to be taken into consideration. The system design needs to cater for cooling as well as heating according to the temperature conditions; the switchover being automatic, based on the requirements. The design is, therefore, based on the following general requirements.

(i) The specified exterior and interior conditions of air temperature and humidity,

(ii) The thermal characteristics of the coach,

(iii) Solar radiation,

(iv) The number of passengers,

(v) The ventilation air required for the passengers,

(vi) Interior heat generating appliances such as fans, motors, lights, heating devices used, etc.

From the design calculations for cooling of the coach the required cooling capacity (K cal/hr) and the ratio of sensible heat to total heat (SHF) are worked out which in turn determine the apparatus dew point (ADP), the temperature of the air leaving the evaporator, the temperature rise of the air required to maintain the interior conditions and the total air flow needed to absorb the sensible and latent heat loads.

The family of curves based on a design study for the coach requirements is shown in figure 1. These also show the effect of varying the interior relative humidity within the normally permitted limits of passenger comfort. It can be seen that the

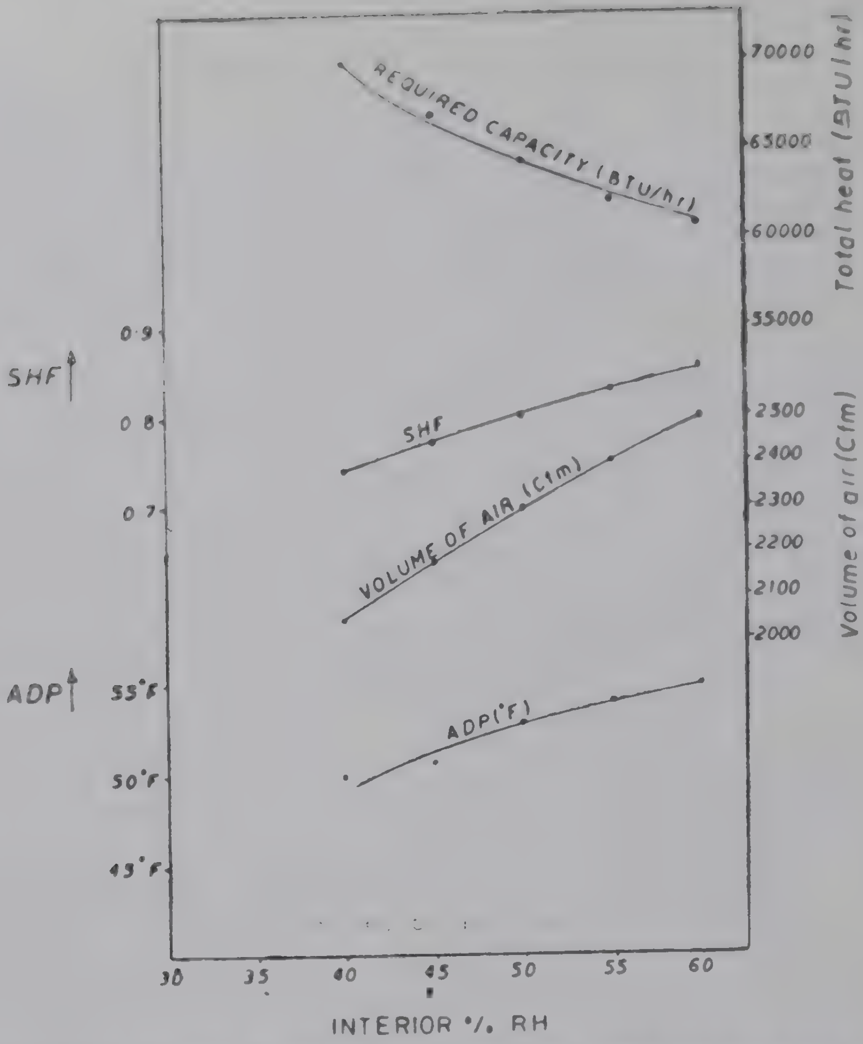


Fig. 1. **Cooling capacity, sensible heat to total heat, dew point and air-flow curves**

equipment capacity, SHF, ADP and air flow are all intimately related and that the respective values become unalterably fixed in relation to the selected value of interior humidity.

It is interesting to note that if the air mass flow is deliberately restricted, for any reason, the internal relative humidity will fall, as will the ADP and the cooling capacity must increase to meet the additional load required for this increased dehumidification. Furthermore, the compressor saturated suction temperature must fall (since it is related to ADP) and the pumping rate will be reduced ; therefore a larger displacement compressor may be required because the required capacity is more and available capacity is less at lower suction pressure.

In actual practice such precise working conditions cannot be always obtained as the design is based on the limiting atmospheric condition in the hottest region and the full passenger loading. For major portion of the period such conditions do not prevail and therefore for partial load conditions the required degree and form of capacity control is determined to ensure equally satisfactory operation at such load keeping in view the varying operating conditions not only from season to season but at times within a single journey from north to south. The coaches are designed for

comfort air-conditioning both for the summer and winter, to maintain the following conditions:

Outside temperature dry bulb °C	45	35	25	4
Outside relative humidity %	25	50	92	
Inside temperature dry bulb °C	26	23	22	20

The comparative results of the heat load calculations to meet the requirements of comfort conditions mentioned above for the three types of air-conditioned coaches proposed to be built are indicated in the Table 1.

TABLE 1: *Heat load calculations*

Type of coach	Composite First + chair	Chair Car	Two tier sleeper
Passenger occupation	10+ 34 BTU/hr	71 BTU/hr	48 BTU/hr
Heat gain from:			
(i) Conduction and solar radiation	12497	12497	12497
(ii) Electrical appliances lighting etc.	4225	5805	4093
(iii) Passengers	17600	28400	19200
(iv) Ventilation*	35800	46950	31410
(v) 10% extra for infiltration	7012	9365	6720
Total	77134	103017	73920
Refrigeration capacity (Tons)	6.4	8.6	6.2

Note: Air requirements are based on the figures of 25cfm/person for first class where smoking is permitted and 12.5 cfm/person for chair car where smoking is prohibited.

Air-conditioning Equipment

Air-conditioning equipment of suitable capacity operating on the electro-mechanical system of refrigeration employing Freon 12 or its equivalent as the refrigerant is used on the coaches. To ensure reliability and to permit operational flexibility, the design provides for two sets of equipment so that in case of failure of any one of the equipment, the system does not fail totally. The electrically driven open type compressor condenser unit is mounted on the underframe while the air-conditioning unit comprising of the blowers for air handling the cooling coil and the heating element is mounted in the roof of the coach near the entrance passage.

The conditioned air is circulated throughout the coach by means of two sets of centrifugal fans at both ends. These fans discharge the air into ducting above the interior ceiling from which it is distributed through numerous discharge apertures. Usually about 75% of the total air is recirculated. The return air passes through a system of filters while the fresh air is drawn from outside the coach through viscous oil filters.

Power Supply

Electric power supply arrangements on railway coaches have their own peculiar problems. In the fully air-conditioned train like the Rajdhani and Deluxe trains, power is supplied at 400V 3 phase 50 Hz by making use of the 'End-on-

generation' system in which diesel generating sets of appropriate capacity are installed in the coaches at both ends of the train composition from where power is distributed throughout the train through train line cables, interlocked couplers and suitable transformers to step down the voltage to appropriate levels for utilization in coaches. On self-contained coaches of the type indicated, the power required is about 35 to 40 kW which can be obtained either by provision of an underslung horizontal diesel generating set or through generators driven through suitable arrangement of drive from the coach axle.

Use of underslung diesel generators present the following problems :

(i) Manufacturing capacity of horizontal high speed diesel engines of the requisite capacity in the country is not available at present.

(ii) The generating sets and their accessories occupy considerable space on the underframe which is conjested by air-conditioning equipment.

(iii) Maintenance of the diesel engines in restricted space below becomes difficult.

(iv) As the air-conditioned coaches are located in the central position of the train composition, underslung diesel sets will create excessive noise and will be a nuisance to passengers of adjacent non-air-conditioned coaches as well as on platforms.

(v) Potential fire hazard.

(vi) Acute shortage of diesel oil.

They, however, will have an advantage of generating power at the industrial frequency and supply voltage to enable use of standard electrical equipment. Considering the disadvantages, however, the use of underslung diesel generating sets for power supply has been ruled out at least for the present.

Axle generation has been universally adopted on the railways for small power requirement of lighting and ventilation. Use of indigenous equipment of about 40 kW capacity has, however, so far not been made and development of appropriate capacity generators is essential. The generation being restricted during periods of runs above the cut in speed only, a self-contained coach needs a power supply through batteries when it is standing. Use of dc power supply therefore, becomes inevitable for which the standard voltage of 110V is selected. Batteries add considerably to the coach weight apart from occupying underframe space. Development of about 18 to 20 kW variable speed brushless alternators with associated regulating and control equipment to supply a steady power supply at 110V dc has been undertaken. Two such generators mounted on the bogie transom between the wheels driven by V-belt off the coach axles will be used for providing the necessary power. Even though axle generation is conveniently possible, several operating constraints apply to this system which have to be met in actual service. The constraints under which such services can be maintained are :

(i) Self contained coaches of this type can work only on fast trains where the ratio of generation to non-generation period exceeds 3.

(ii) Long halts for any reasons cannot be permitted for such trains. In the event of halts longer than 40 to 50 minutes effective air-conditioning is not possible due to limitation of battery capacity.

(iii) Axle generation imposes a drag on the train equivalent to an addition of about 32 tons of extra coach load to be hauled by the engine at speeds of about 100 to 120 km. Train load therefore gets restricted if more coaches of this type are used on trains.

(iv) Catering services of the kind provided on the Rajdhani express train may not be fully possible on such coaches as use of electrical appliances is not feasible due to limitation of power availability; however service to the extent available from adjacent dining cars and from refreshment rooms at stations could be arranged.

Design of self-contained coaches has, therefore, been undertaken with these restrictions which in normal service can be generally met. The power supply system has been designed making use of the system of axle generation which is a most convenient system for self-contained coaches providing immense fllexibility of operation.

The design of power supply scheme also provides for obtaining external power from 400V 50 Hz 3 phase mains for preconditioning the coaches. The power is taken through a set of power plugs and sockets, rectifying the feed through an appropriate capacity silicon rectifier unit. With this external supply it is possible to provide power for precooling, preheating, lighting or battery charging as the case may be while the coach is standing on a railway siding or at the maintenance depot.

Control Gear

The control equipment for the air-conditioned coaches which is generally mounted in a cubicle near the luggage compartment by the side of the doorway can be divided into two groups viz, controls for the power supply and those for the air-conditioning equipment.

The control and protection of the axle generated power is achieved by the regulating equipment provided for the variable speed alternator. The regulating equipment maintains a constant voltage at a set value subject to the current being within the maximum designed limit at all train speeds above the cut-in speed which is about 25 kmph. A current limiting feature prevents overloading of the machine by reducing the voltage as soon as the maximum set current limit is reached. These two features provide built in safeguards against overload. The system also incorporates a battery charging control scheme to ensure a charging rate at a satisfactory level to obtain a reasonably long battery life. For the protection of electrical equipment, fuses of appropriate ratings are used in the various circuits.

The control gear for preconditioning supply incorporates the preconditioning accessories including power plugs and sockets rectifying equipment for obtaining dc supply and associated switching and protective equipment.

The control gear for the air-conditioning equipment comprises of contactor type switchgear for the control of compressor unit, condenser unit, the air handling unit, and the heater units. The temperature conditions within the coach are

thermostatically controlled. Three different temperature settings for heating and cooling are provided which can be selected by manually operated control switches. The transition from operation of cooling to heating equipment or vice versa is automatically effected through a system of pilot relays which sense the temperature conditions with the help of extremely sensitive thermostats. The controls for heating and cooling have to be very precise, as the coaches operate on a limited power supply arrangement and unnecessary use of energy has to be avoided at all costs. For the protection of the refrigeration systems low pressure cutout and high pressure cutout switches are provided to ensure the safety of the compressors and associated equipment.

Future Trends

The demand for air-conditioned travel is growing rapidly all over the world. Even in Western countries where climatic conditions do not warrant cooling for most of the period, air-conditioning is finding increasing acceptance; In a hot country like ours the growth of the air-conditioned coaches is therefore, inevitable. Indian Railways are now embarking on the scheme to build about 100 self-contained air-conditioned coaches of high passenger carrying capacity. Further additions, however, would largely depend upon the popularity of these designs and future trends in the development of air-conditioning plants.

Majority of railway passenger coaches all over the world, fitted with air-conditioning systems use as a source of refrigeration, a central plant operating on the principle of vapour compression system of refrigeration. The following unconventional systems have also been tried out on a few foreign railway systems :

(i) Peltier system—air-conditioning system for heating and cooling based on the Peltier effect.

(ii) Individual cooling units of the commercial type for each compartment.

(iii) Compressed air cooling system.

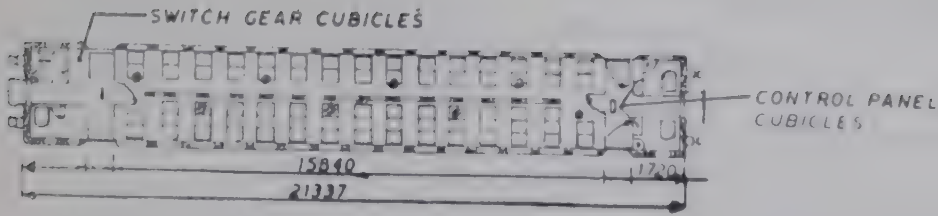
(iv) Heat driven vapour cooling systems.

Although experimental work on these systems had been done on some railways, none of these have so far proved to be more efficient and economical compared to the present conventional electromechanical system.

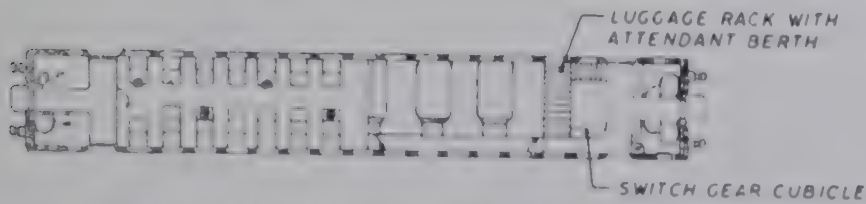
There can, however, be a good possibility of adopting individual cooling units mounted in the roof of a coach. Use of sealed and semi-sealed air-conditioning units on large scale has not been possible on the Indian Railways due to non-availability of proved, reliable equipment. Development of such equipment in future to reduce the growing maintenance cost is essential. Compact designs with such units would also enable utilization of the available floor space more effectively. Railways would, therefore, be looking forward to our refrigeration industry to evolve suitable designs to meet these requirements.

Conclusion

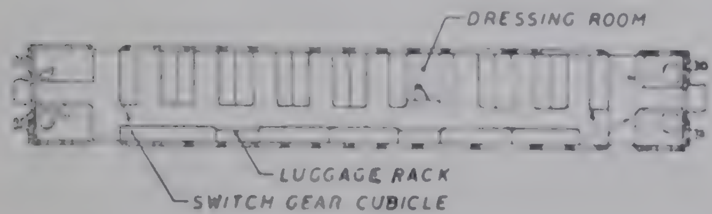
Conventional systems with open type electromechanical system is now being used almost exclusively on railway coaches in India. The need to evolve systems



A C CHAIR CAR



A C FIRST CLASS SLEEPER CUM CHAIR CAR



A C 2-TIER SLEEPER

which are economical in cost, power requirements and energy consumption and day-to-day maintenance, however, is felt acutely. Only then will it be possible to achieve significant improvements and to take rapid strides for providing the much needed air-conditioned comfort during the journey at reasonable costs for a large number of passengers on the railways.

Computer Application for Predicting the Performance of Heating and Cooling Coils

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Introduction

Ever since extended surface coils were first introduced for heating and cooling devices, the engineers are facing problems for finding suitable methods for producing the performance of such devices at various loads. Many methods have been suggested by various authors for doing so but most of them are cumbersome, time consuming and need particular experimental testing. Making these tests and comparing with standards takes a lot of time. Results given by various authors involve

a large number of variables and complicate analysis, which are of no practical use. The subject of this paper is to find the yardstick which may directly give us the performance within reasonable accuracy, and to computerize the various mathematical relations as suggested by various authors in terms of variable input data conditions and to get the variables in terms of tabular form instead of graphs so that the selection of devices become simple and reliable.

Computational Procedure

The 'effectiveness contact method' 1, the 'surface temperature method' 2 and the 'Butt method' 3 are used for the calculations carried out hereunder. The working equations are given in the cited references 1-3. A graph is plotted (Fig. 1) with the reciprocal of coil multiplier (1/M) as the abscissa and the ratio of effectiveness coefficients and the contact factor as ordinates. The flow charts for the computer programmes developed using the above methods are given in Figs. 2 to 4.

Illustration

1. *Effective contact method :* Let us assume exhaust Cfm $Q_A = 80$; evaporator Cfm $= 350$; No. of rows deep $= 4$; inner dia $= 0.43''$; outer dia $= 0.5''$; As = the ratio of outside area to the face or row area $= 120$; CF = contact factor $= 0.999$; t_1 = room air temperature $= 78°F$; $\triangle T = 40°F$.

From Graph $M = 1.66$ $\therefore Q_1 = 96.8$

Cooling capacity $= Q_1 \times T = 96.8 \times 40 = 3872$ BThU/hr.

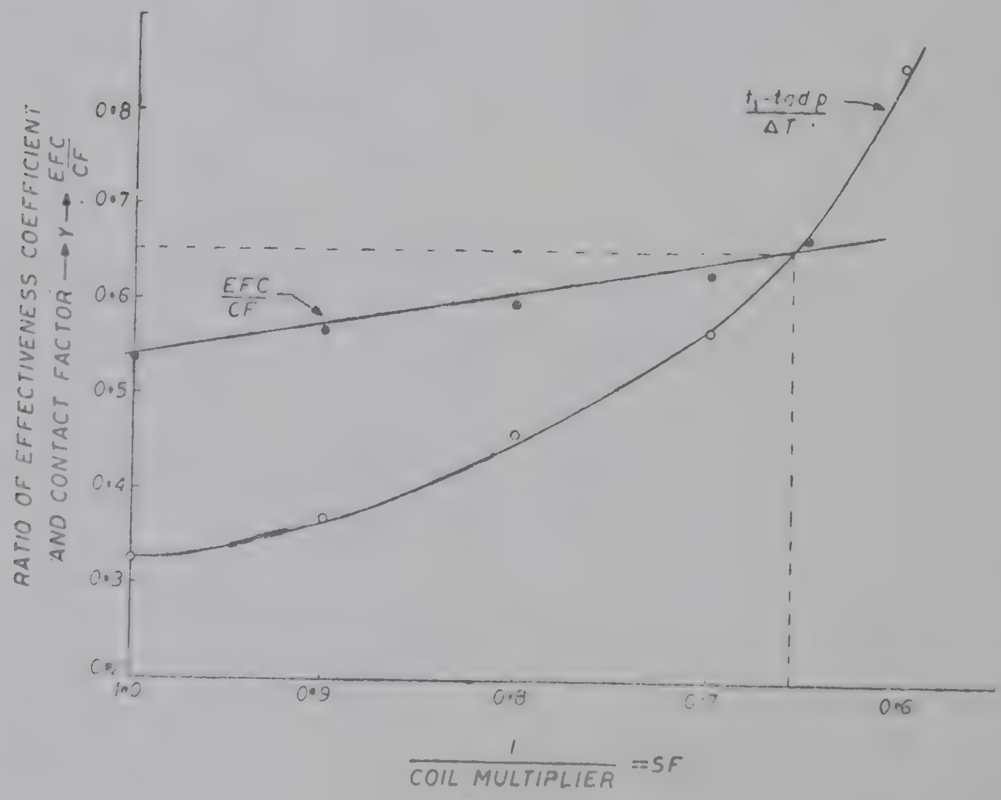


Fig. 1

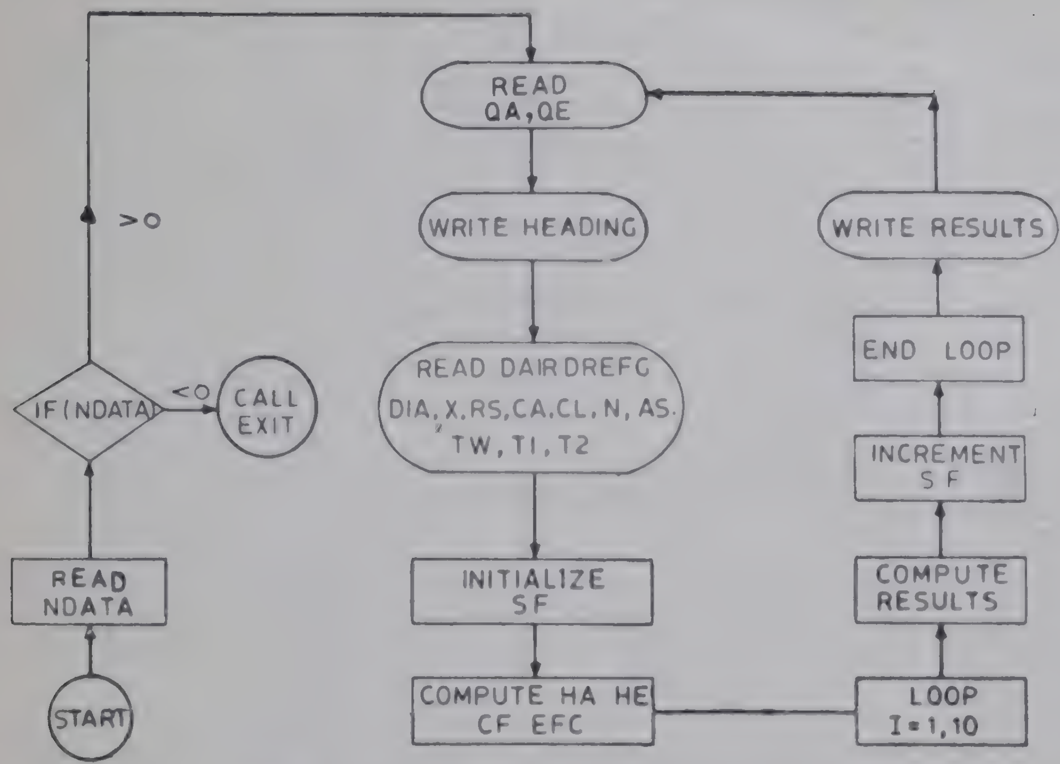


Fig. 2. Flow Chart for effectiveness method

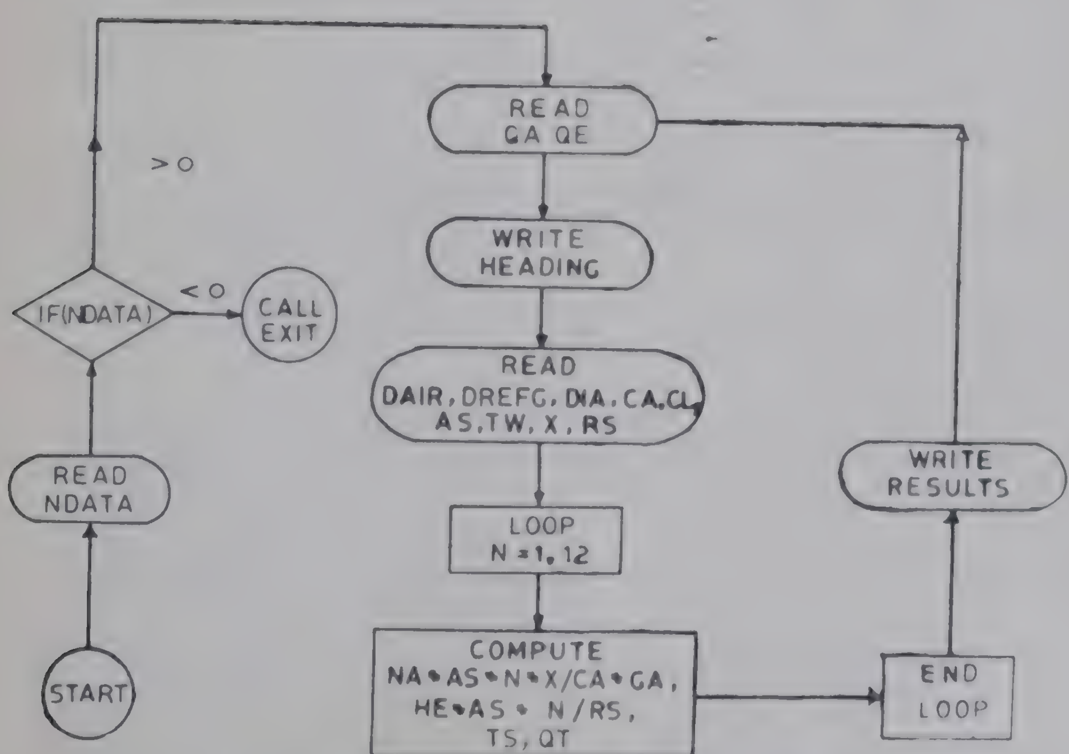


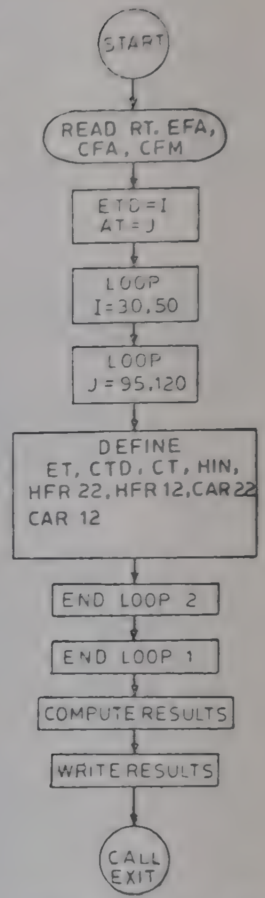
Fig. 3. Flow Chart for surface contract method

2. *Surface temperature method:* Given Q_A = 80 Cfm, Q_E = 350 cfm, Nominal dia of the tube = 3/8″; inner dia = 0.43″; outer dia = 0.5″; C_a=0.243 BThu/lb°F; C_i=0.306 BThu/lb°F; No. of rows deep=2; ratio of outside surface area to the face area i.e. A_s=10; room temperature t_1=78°F; ambient air dry bulb temperature t_2=95°F; Heat transfer ratio of outside the inside surface=1.3; efficiency of the fin (X)=80%; refrigerant temperature=47°F.

Result: Surface temperature t_s = 52.5°F,
Coil capacity Q_t=7865 BTU/hr.

3. *R. W. Butt Method:* For evaporator temperature difference ETD=38°F, evaporator temperature of 40°F, condenser temperature difference CTD=30.77°F, EFA=evaporator fin area=90 sq.ft.; CFA=condenser fin area=150 sq.ft.; cfm= 2.54; Heat in=ETD×3.0×EFA.

AT °F	CT 0°F	Heat pumped for F - 12	Cooling capacity BTU/hr.	Power consumption kW
95.0	125.77	10702.81	10481.40	1.0970
96.0	126.77	10618.39	10439.19	1.0883
97.0	127.77	10535.29	10397.64	1.0798
98.0	128.77	10453.48	10356.74	1.0714
99.0	129.77	10372.94	10316.47	1.0632
100.0	130.77	10293.62	10276.81	1.0550
101.0	131.77	10215.51	10237.75	1.0470



Flow for R. W. Butt method

Now at AT=101°F, power consumption=1.047 kw,

$$\text{Cooling capacity} = \frac{10237.75 + 10215.51}{2} = 10226.63 \text{ BThu/hr.}$$

This value is nearly the same as calculated from ASHARE Data Book.

Relative study of methods

Effectiveness contact method	Surface temperature method	Ambient air and evaporator temperature diff. method
This method can be used for wet as well dry coil (M=1) so it can be used for heating as well as cooling devices for any refrigerant, air, water, etc.	This method deals with dew point temp. of a room so can be used only for demudified coils like water coolers, air conditioners.	It deals only with air conditioners.
Since it illustrates the effectiveness coefficient factor with respect to various parameters, it can be used for any type of coil at any load condition.	Can be used for any number of coils in row at any load conditions.	Can be used for any type of air conditioner only by replacing cfm by $\frac{\text{(New cfm)}}{2.54}$ Compressor cfm
Various variable and efficiency factors are to be multiplied for getting correct result.	The method already has accounted all variables and give direct results.	It has already accounted losses in empirical formula and gives direct result.
Can be used designing air conditioner tube, heating tubes etc.	Can be used for design tubes and number of rows deep.	Can be used for checking efficiency of compressor at various load and temp. conditions.
Here calculation and curves are to be plotted for getting operating range.	It gives direct result if A_s is known.	It gives direct result, no calculation.

Conclusion

By observing the results and fact which have been achieved, one come to conclusion that method (1) (Effectiveness Contact Method) is reliable and accurate. It considers all non-dimensional factor but it is most suitable for heavy load coils.

Ambient air temperature method is also reliable, simple and can be used for standardization of air conditioners at various load, temperature conditions. It gives all the results at hand and does not need any complex calculation.

Therefore one can use all the three methods at any condition and at any place to predict the performance of various heating and cooling devices.

Nomenclature

A_s	$= \dfrac{\text{sq ft of outside area}}{\text{sq ft of face area/row}}$
CF	$=$ Contact factor $= 1 -$ Bypass factor $= 1 - \exp\left(-\dfrac{US}{W_a C_a}\right)$
C_a	$=$ Specific heat of air $= .243$ Btu/lb°F
C_i	$=$ Inner fluid specific heat, Btu/lb°F
CTD	$=$ Condenser temperature difference
C_A	$= \dfrac{\text{Mass flow rate of air}}{\text{Cross section area of tube}} = \dfrac{\text{lb}}{\text{hr ft}^2}$
GE	$=$ Mass flow rate of refrigerant lb/hr ft^2
HA	$=$ Air side heat transfer coefficient Btu/hr ft^2 °F
HE	$=$ Refrigerant heat transfer coefficient Btu/hr ft^2 °F
h_1	$=$ Enthalpy of air (BTU/lb) entrance flow side.
h_2	$=$ Enthalpy of air BTU/lb (exit flow side)
M	$=$ Coil Multiplier $= (h_1 - h_2)/C_a(t_1 - t_s) = \dfrac{1}{SF}$
N	$=$ Number of rows deep
Q_A	$=$ Exhaust cfm
Q_E	$=$ Evaporator cfm
RT	$=$ Room temperature (°F)
R	$= \left(\dfrac{W_a C_a}{W_1 C_1}\right) M$ Non-dimensional factor
RS	$=$ Heat transfer ratio $= \dfrac{\text{outside surface}}{\text{inside surface}}$
S	$=$ Coil total heat transfer surface area ft^2
t_w	$=$ Average temperature of refrigerant
t_{adp}	$=$ Apparatus dew point temperature of the coil °F
t_1	$=$ Room temperature (dry) °F
t_2	$=$ Air ambient temperature (dry) °F
t_r	$=$ Refrigerant temperature
ΔT	$=$ Temperature difference between entering air and inlet refrigerant temperature.
U	$=$ Overall heat transfer coefficient $= \dfrac{RS}{HS} + \dfrac{1-X}{XHA} + \dfrac{1}{HA}$
V_w	$=$ Average velocity of refrigerant $=$ cfm/internal cross sectional area
W_1	$=$ lb/hr refrigerant
X	$=$ Efficiency of fin.

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Cascaded Desert Coolers

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Introduction

Considerable difference in dry and wet bulb temperature of air in desert area during summer makes the desert-cooler a useful device for lowering the air temperature by simple humidification process. But the condition of the air coming out of the desert-cooler is not good for human comfort because of its high relative humidity. If in a heat exchanger the humidified air at low temperature from the desert-cooler can be utilized to cool another stream of air which is taken from the atmosphere and the amoun tof humidified air is large as compared to that of outside air, then the dry bulb temperature of the outside air (hereafter referred to as cooled air) will be reduced up to that of humidified air. Since no moisture is being added to the cooled air its humidity will remain constant and its relative humidity will be lower than that of humidified air. Therefore the cooled air will have lower wet and dry bulb temperatures than those of atmospheric air. If this cooled air is further humidified, it will result in lower effective temperature than that of the original atmospheric air if humidified. Hence cooled air after humidification will be more comfortable than the air coming out of an ordinary desert-cooler.

Therefore there is an urgent need to investigate the practical validity of this idea. Since the humidification is done at two places a name Cascaded Desert-Cooler will be used for this apparatus.

In the present work, the humidification of atmospheric air and the exchange of heat between humidified and cooled air takes place simultaneously. One stream of air flows over a metal plate on which a wet wick is placed hence this stream will get humidified. At the same time another stream of air flows below the metal plate this stream will be cooled because of heat transfer from it to the interface i.e., between wick and humidified air through the metal plate and wick.

Analysis

Fig. 1 shows a stream of air flowing over a metal plate over which a wet wick is placed. If the length of the metal plate and wick is sufficient enough, the relative humidity of the humidified air coming out of the heat exchanger will be 100% and it will follow a path AB on a psychrometric chart Fig. 2. Another stream of air which flows below the metal plate will be cooled along the path AC as no moisture is being added to this. If the efficiency of the heat exchanger is 100% then the temperature of the two streams should be the same as they come out of the heat exchanger. The dry bulb temperatures of humidified and cooled air and the interface temperature are expected to vary along the length of the heat exchanger as shown in Fig. 3. Since the

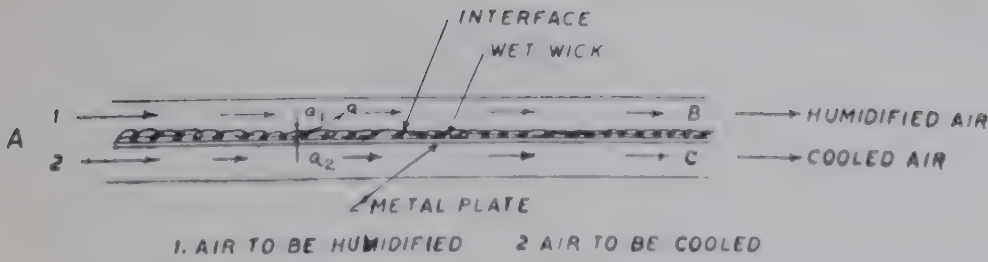

Fig. 1. Heat exchanger showing metal plate and wick

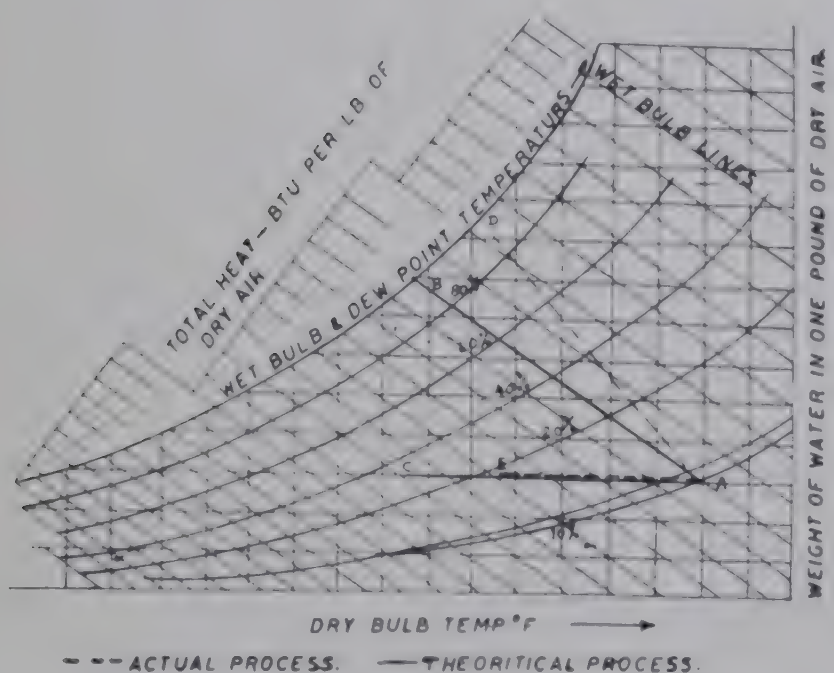

Fig. 2. A—Condition of Outside Air; B—Theoretical Condition of Humidified Air; C—Theoretical Condition of Cooled Air; D—Actual condition of Humidified Air and E—Actual Condition of Cooled Air

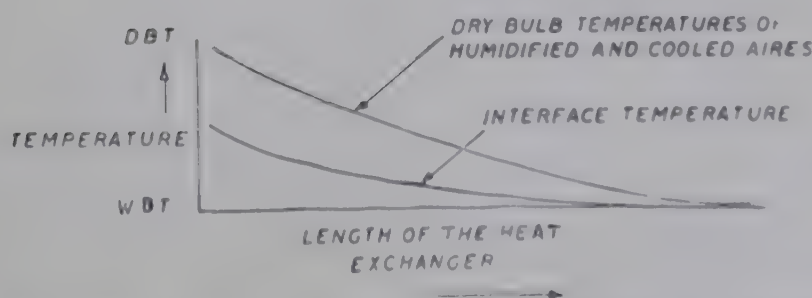

Fig. 3. Variation in Dry Bulb temperatures of Humidified and Cooled Air and interface temperature along the length of the heat exchanger

temperatures of humidified air stream and cooled air stream are always higher than the interface temperature, the sensible heat of both the streams will flow towards the interface as shown in Fig. 1. Let these quantities of heat be Q_1 and Q_2. The sum of these two quantities must be sufficient enough to supply the heat of evaporation to the water at the interface. Let this heat of evaporation be Q. Then the heat balance can be written as

$$Q=Q_1+Q_2 \tag{1}$$

Q_1 is given by heat conversion law

$$\frac{Q_1}{A_1}=h\,(t_1-t_i) \tag{2}$$

h i.e. the convective heat transfer coefficient is given by

$$\frac{hD}{K} = 2.23\,(R_e)^{0.8}\,(P_r)^{0.3} \tag{3}$$

$$D = \text{hydraulic diameter} = \frac{4\,ab}{a+b}$$

$$R_e=\frac{Vd\rho}{\mu}\ ;\ P_r=\left(\frac{\mu c_p}{k}\right)$$

Q_2 = heat flowing towards the interface of area Al from the air to be cooled through the metal plate and wick can be calculated from the following relation

$$\frac{Q_2}{A_1}=U\,(t_2-t_1) \tag{4}$$

$$U=\frac{1}{\left(\frac{1}{h_2}+\frac{t_w}{k_w}+\frac{t_e}{k_e}\right)} \tag{5}$$

h_2 can be calculated using equation (3).

Quantity Q is the heat required to evaporate water from interface of area A_1.

$$\therefore\ \frac{Q}{A_1}=\frac{W}{A_1}\ h_{fg} \tag{6}$$

$$\frac{W}{A_1} = \text{weight of the water evaporated/unit area}$$

$$= h_d\,(c_i-c_\infty) \tag{7}$$

The mass transfer coefficient h_d is given by

$$h_D=\frac{D_m}{D}\ (Re)^{0.83}\ (N_s)^{0.33}x0.033 \tag{8}$$

$$N_s = \frac{\mu}{\rho D_m}$$

$$D_m = \frac{0.000146}{PT}\times\left(\frac{T^{2.5}}{T+441}\right) \tag{9}$$

In the above equations all other quantities are known except the interface temperature which can be determined by trial and error method.

Determination of length of heat exchanger

The interface temperature is not constant along the length of heat exchanger. It can be assumed that humidification is taking place in several intervals as shown in Fig. 4. For each interval an interface temperature is calculated by trial and error. For a particular humidification range the interface temperature is assumed constant and length of heat exchanger required for that particular range is calculated. Total length is calculated by adding the lengths required for each humidification range.

The length of the heat exchanger required for the apparatus was calculated under following assumptions.

Air velocity = 900 fpm in both the passes

Outside air D_{bt} = 105°F, W_{bt} = 65°F

The relative humidity of the *humidified air* coming out of heat exchanger = 90%

The length is calculated by the following relations

$$mc_p \triangle T = U.A.\ (\triangle T)_{ln} \tag{10}$$

$$A = w \times L = L \text{ for unit width}$$

$$L = \frac{mc_p \triangle T}{U.(\triangle T)_{ln}} \tag{11}$$

The required length of the heat exchanger = 4 feet.

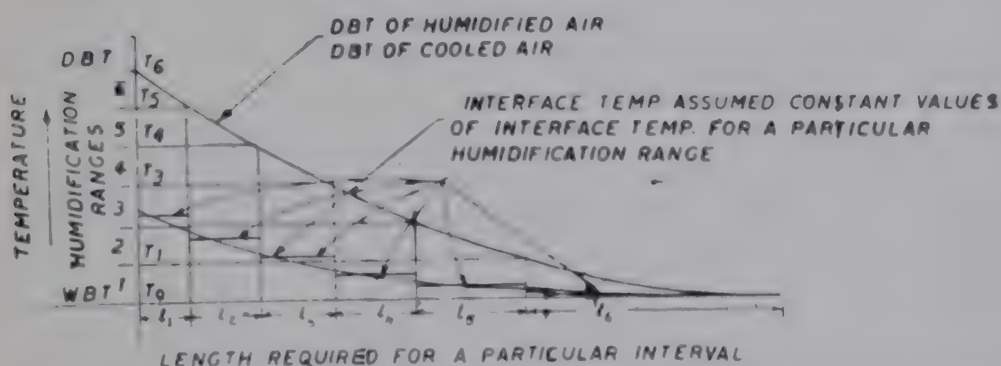

Fig. 4. Various humidification range and the assumed constant values of interface temperatures for each range

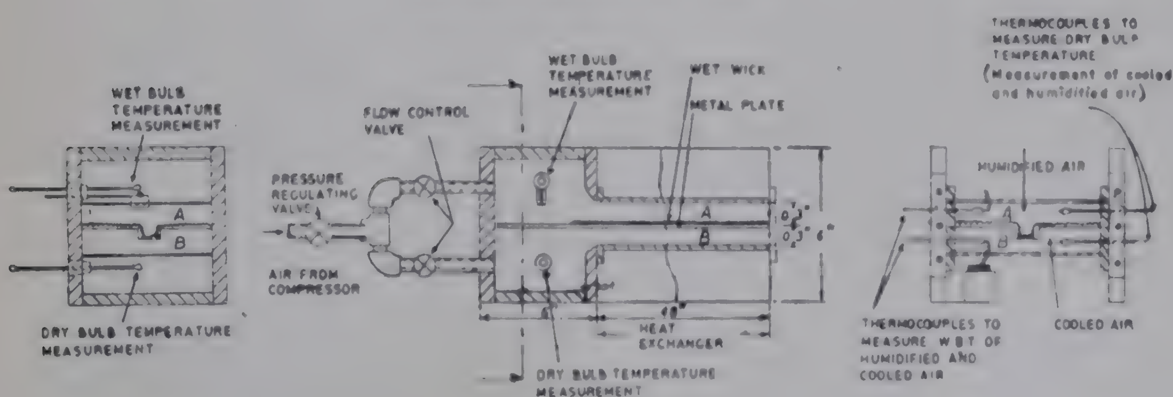

Fig. 5. General set up

Experimental Programme

The general arrangement of set up for conducting test is shown in Fig. 5. The apparatus consists of two passes A and B separated by a metal plate over which a wet wick is placed. A small channel at the middle of the plate is made to take care of make-up water which should be supplied to keep the wick wet. At the exit end of the apparatus a small quantity of water is poured inside the channel from time to time. The supply of air for the apparatus is taken from the compressor.

The following observations are made during the experiment : The wet bulb temperature of the incoming air is measured in the pass A and dry bulb temperature in the pass B. These measurements are taken in the portion just prior to the heat exchanger as in Fig. 5.

Dry and wet bulb temperatures of outgoing air streams (humidified air and cooled air) are measured separately by thermocouples as shown in the right side view Fig. 5.

The average velocity of air in a particular pass is measured by Pitot tube.

All the valves of the apparatus were completely opened before starting air supply from compressor. Then the supply valve of the compressor was slightly opened. At the same time sufficient amount of water was poured inside the channel such that the wick becomes wet. Air velocity was made 900 fpm in both the passes by flow control valve and then observations were taken after a steady state was reached. The experiment was repeated on various days to test the effectiveness of apparatus under different conditions.

These observations are shown in Table 1. Column (8) gives the theoretical decrease in the wet bulb temperature of cooled air (i.e. W_{bt} corresponding to condition A minus W_{bt} corresponding to condition C as shown in Fig. 2). Column (9) and (10) were obtained from Psychrometric chart, knowing the conditions of atmospheric air, cooled air and humidified air. Column (9) gives the amount by which the humidified air is heated from its original condition. Similarly column (10) gives the amount by which cooled air is cooled from its original condition.

It is clear from the table that the wet bulb temperature of the cooler air is not dropped up to the theoretically predicted extent, because the theoretical decrease is predicted assuming adiabatic humidification in the heat exchanged. The actual process is shown by the dotted line AD in Fig. 2. The humidified air can cool the cooled air only upto the point E. The amount of heat by which humidified air is heated is more than the amount by which the cooled air is cooled. This is due to the inefficient working of the heat exchanger.

Using cascaded type cooling the wet bulb temperature of outside can be decreased by 8 to 11°F in dry hot weather.

Cooled air from heat exchanger, if further humidified, will be comfortable to the human body. This type of desert cooler will be cheaper than the air conditioner which the common man cannot afford.

Condition of outside air		Condition of humidified air		Condition of cooled air		Reduction in wet bulb temperature of cooled air		Heat exchanged in the heat exchanger	
DBT °F	WBT °F	DBT °F	WBT °F	DBT °F	WBT °F	Experimental °F	Theoretical °F	Amount by which pass A air, is BTU	Amount by which pass B air is cooled BTU
1	2	3	4	5	6	7	8	9	10
98.2	62.0	71	70	72	53	9.0	14.5	6.25	5.85
101.0	63.0	73	71.5	75	54	9.0	15.0	6.70	6.00
69.8	53.0	58	58	57.5	48	5.0	7.5	3.10	2.85
105.0	66.0	76	74	80	57	9	14.5	6.80	6.40
108.0	67.5	78	76	79	57.5	10.0	14.5	7.60	7.20
109.8	66.5	79	76	80	56.5	10.0	16.0	8.40	7.05
109.2	67.0	79	76	80	57	10.0	16.5	7.95	7.20
110.5	69.5	80	78	82	60.0	9.5	14.0	7.95	7.20
82.0	57.0	65	64	66	50.5	6.5	10.5	4.85	3.90
108 0	66.3	75.5	74	78.5	55.5	11.0	16.0	6.45	7.70
109.0	69.0	80	76.5	82	60	9.0	14.0	6.85	6.80
104.6	68.5	79	76.5	80	60.5	8.0	12.5	7.40	5.90
107.5	69.0	80	77	82	60.5	8.5	13.5	7.35	6.45
110.5	69.5	80	78	82	60	9.5	14.0	7.95	7.20
112.0	68.0	81	77.5	84	59	9.0	15.5	8.15	6.70

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Nomenclature

A_1	=Area of interface ft^2
a	=Thickness of a pass—ft
b	=Width of a pass—ft
C_i	=Density of water vapour corresponding to interface temperature lbm/ft^3
C_p	=Specific heat of air BTU/lb °F
C_∞	=Density of water vapour present in upper stream air at a particular temperature. lb m/ft^3
D	=Hydraulic diameter—ft
D_m	=Diffusion coefficient—ft^2/hr
h^1	=Convective heat transfer coefficient on wick side. BTU/hr °F ft^2
h^2	=Convective heat transfer coefficient on cooled air side BTU/hr °F ft^2
h_d	=Mass transfer coefficient—ft/hr
h_{fg}	=Latent heat of water corresponding to interface temperature. BTU/lb m
k	=Thermal conductivity of air—BTU/hr ft °F
k_o	=Thermal conductivity of metal sheet—BTU/hr ft °F
k_w	=Thermal conductivity of wick—BTU/hr ft °F
L	=Length of the heat exchanger=ft
N_s	=Schmidt Number
P_r	=Prandtt number
Q	=Heat of evaporation for water evaporating at the interface of area A_1—BTU/hr.

Q_1	= Heat flowing towards interface of area A_1 from upper stream air—BTU/hr.
Q_2	= Heat flowing towards interface of area A_1 from cooled air—BTU/hr.
R_e	= Reynauld Number
t_o	= Metal thickness—ft
t_w	= Wick thickness—ft
t_i	= Interface temperature—°F
t_1	= Temperature of upper stream air—°F
t_2	= Temperature of lower stream air—°F
T	= Absolute temperature °F
U	= Overall heat transfer coefficient BTU/hr °F ft^2
V	= Velocity of air fpm
W	= Mass of water vapour evaporated from interface of area A_1—lb m
ΔT	= Decrease in temperature of cooled air in a particular interval—°F
ρ	= Density of air (lb m/ft^3)
μ	= Viscosity of air lbm/ft hr.

Use of Cold for Process and Comfort in an Integrated Steel Plant

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Introduction

For process and comfort needs cold is used extensively in an integrated steel plant. Keeping pace with the international development of steel technology integrated steel plants of 2·3 million tonnes initial capacity have been suggested. For a steel plant of such size the requirement of cold is 8·10 million kilo-calories/Hour.

Cold is used for air-conditioning of control rooms, pulpits, instrument rooms, premises of laboratories and production control centres. Cold is used for process needs such as cooling of electrical machines, cooling of coke oven gas for recovery of naphthalene, cooling of benzol vapour and acetylene cylinders. Cold is also utilised for providing air-conditioned storage of electrical spares and tar bonded dolomite bricks

Choice of Methods of Producing Cold

In modern air-conditioning installations three types of refrigerating machinery are used for producing cold, namely vapour compression, absorption and steam-jet.

The disposition of consumers of cold is such that chilled water system is the only solution for air-conditioning needs. For process needs we require chilled water. Adjacent to the main shops such as Coke ovens and By-products plant, Blast furnace and Sintering plant, Steel Melting Shop, Rolling Mills—chilled water plants are put up to cater to a group of consumers. Alternatively, a centralised chilled water plant can be put up to serve all the consumers at different units of the steel plant.

In a steel plant there are plenty of secondary energy resources which are utilised to produce steam. Steam is generated by recovering heat of hot exhaust gases from a furnace or a combustion process (Waste heat boilers). Steam is generated on the principle of evaporative cooling while cooling furnace elements or equipment in very high temperature zones. By evaporative cooling of blast furnace shell only, on an average, steam produced may be as high as 15 KG/hr. per 1 M^3 of useful volume of blast furnace. Due to the availability of steam and simplicity of technological process, steam jet refrigeration system has been suggested as the most suitable method of producing cold. Apart from steam, bulk quantity of water is required for condenser cooling which is available at a cheap rate in a steel plant.

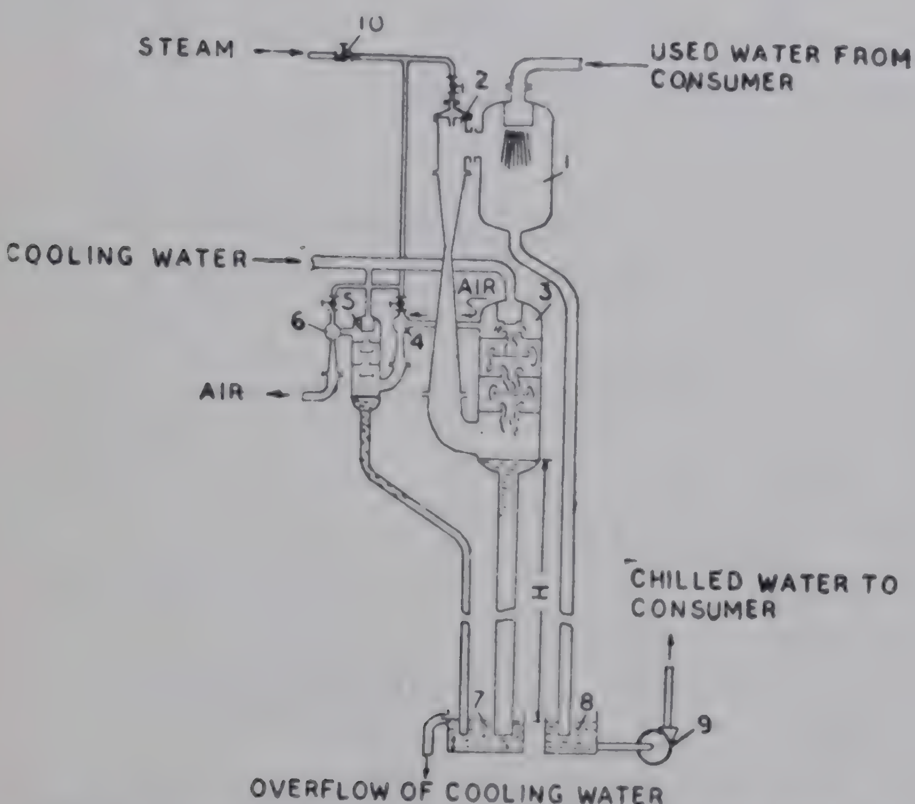

FIG. I SCHEME OF STEAM JET REFRIGERATION UNIT WITH BAROMETRIC MIXING TYPE CONDENSER

1. EVAPORATOR
2. MAIN EJECTOR
3. MAIN CONDENSER
4. AUXILIARY EJECTOR—1ST. STAGE
5. AUXILIARY CONDENSER
6. AUXILIARY EJECTOR-2ND. STAGE
7. BAROMETRIC SEAL FOR CONDENSATE & COOLING WATER
8. BAROMETRIC SEAL FOR CHILLED WATER
9. PUMP FOR CHILLED WATER
10. THROTTLE VALVE

Steam-Jet Refrigeration Principle

Principle of working of steam-jet refrigeration system is basically the evaporation of water in vacuum created by steam-ejectors. In figure 1, scheme of steam-jet refrigeration unit has been shown. Water circulating in cooling systems comes to evaporator where vacuum is created with the help of main steam ejectors to secure the boiling of water at the required temperature. For example, at an absolute pressure of 7.513 mm Hg, water boils at +7°C. In the scheme, mixing type barometric condenser has been shown. In the main condenser, vacuum is created by 1st stage auxiliary ejector. Steam from this auxiliary ejector is condensed in another mixing type condenser where vacuum is created by 2nd stage auxiliary ejector discharging into atmosphere. Chilled water from the evaporator is collected in the barometric trough from where it is pumped to the consumers.

Steam-Jet Refrigeration—Steel Plant Application

In India, for the Bhilai and Bokaro Steel Plants, the U.S.S.R. has supplied steam-jet refrigeration units for producing 850 M^3/hr. chilled water at 12°C respectively. The chilled water plants are installed outdoor except the pumps, electrical installation and instruments. Test results of the U.S.S.R. built steam-jet refrigerating machines are shown in the table below:

Regime	Steam pressure in atmospheres (gauge)	Steam consumption kg/hr	Condenser cooling water M^3/hr	Temperature of condenser cooling water °C		Temperature of chilled water °C	Production of cold 10^6 Kcal/hr
				Initial	Final		
1.	2.0	8200	835	37.2	45.1	12.9	1.08
2.	2.05	7400	760	32.0	39.5	17.0	0 63
3.	8.0	7260	795	37.9	45.8	13.3	1.245
4.	6.7	6480	730	32.6	39.4	7.3	0.754

Scheme of Cold Supply

In figure 2, principal scheme of cold supply in a steel plant has been shown. Chilled water at temperature of +7°C is sucked by pump from the reservoir and supplied through pipelines to different conditioners where it takes up heat from the air and the hot water at temperature of +13°C is supplied to thc coolers of electric machines through pipe lines of chilled water supply and here the water temperature rises by another 6°C and it comes back at +19°C temperature to the reservoir in the refrigeration station for recooling. Water is sucked from the reservoir by a pump and supplied to the 1st stage evaporator of the refrigerating machine where it is cooled to +13°C. Then the water flows to the 2nd stage evaporator of the refrigerating machine where the water is cooled to +7°C and is delivered to the reservoir after which the cycle repeats.

From the refrigeration station the chilled water is delivered to consumers through steel pipes with necessary insulation. The pipes are laid in covered underground channels or in tunnels which are provided for other pipe lines.

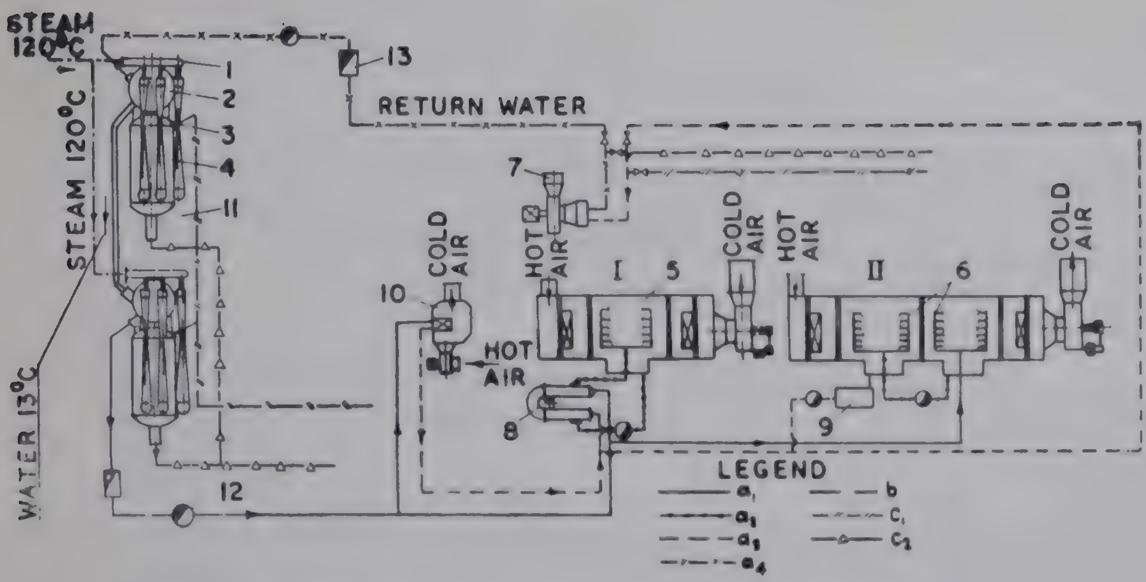

FIG. 2 PRINCIPAL SCHEME OF COLD SUPPLY IN STEEL PLANT

I & II - TWO VARIANTS OF CONDITIONERS WITH SPRAY CHAMBERS AS CONNECTED TO THE SYSTEM OF COLD SUPPLY

1. DISTRIBUTING RING
2. EVAPORATOR
3. EJECTOR
4. CONDENSER
5. SINGLE SPRAY CHAMBER
6. DOUBLE SPRAY CHAMBER
7. INSTALLATION FOR COOLING OF ELECTRIC MACHINE
8. HEAT EXCHANGERS
9. TANK
10. CONDITIONER OF CONTROL POST
11. COOLING INSTALLATION
12. PUMP
13. RESERVOIR

a_1, a_2, a_3 & a_4 — COLD WATER WITH RESPECTIVE TEMP. OF 7, 10, 13 & 19°C
b —— STEAM
c_1 & c_2 —— INDUSTRIAL WATER SUPPLY & RETURN

Chilled water pipe lines are sometimes taken on stockades along with other energy pipe lines.

Conclusion

In a tropical climate like ours demand for cold is very high to provide suitable working conditions. To meet this demand for future steel plants and expansion of existing steel plants, efforts may be made to develop the steam-jet refrigeration plant of commercial size indigenously.

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Application of Evaporative Cooling in Indian Cities A Technical Study

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Introduction

The human body converts the chemical energy of food into work and heat. The heat generated depends upon the body activity. A complex regulating mechanism in the human body keeps the temperature at 27°C. The heat generated in the body must be dissipated to the atmosphere in order to maintain a constant body temperature at 37°C.

Effective temperature (ET) is defined as the index which correlates the combined effects of air temperature, humidity and movement upon human comfort. Mookerjee and Sharma have suggested the effective temperature for an average Indian as :

(i) Comfortable pleasant (lower level) – 21.1°C to 24.2°C (70°F to 75.5°F)
(ii) Comfortable pleasant (upper level)—24.4°C to 26.6°C (76°F to 79.9°F)
(iii) Warm and unpleasant —26.7°C to 28.3°C (80°F to 82.9°F)

A person feels comfortable if he is exposed to an atmosphere which is air conditioned. Air conditioning amounts to heating of the atmospheric air in winter and cooling it in summer and maintaining proper humidity, cleanliness and distribution. Here in this paper Air cooling is discussed.

The two general methods of cooling are refrigerated air conditioning and evaporative cooling. Though air conditioning provides a very accurate control over the temperature and humidity of air, the initial cost is almost 4 times and running cost 6 to 8 times greater than an evaporative cooling plant. In our country there is a great need for conserving power. When a precise control over air properties is not essential, evaporative cooling can be preferred to air conditioning.

In case of spaces cooled by evaporative cooling, the circulation of air within the space is of the order of 30m/min. to 60m/min. considering air circulation, the indoor design conditions for India must be assumed to be 30°C (86°F) and 70% R. H. which corresponds to 26.6°C ET. As the process of evaporative cooling results in increase in humidity, the indoor design conditions for such a system has been taken for the upper comfortable limit of relative humidity.

If unsaturated air is exposed to free and colder water in adiabatic conditions, some of the sensible heat of air is transferred to water and is converted to its latent heat evaporating some of water. This additional vapour increases the humidity of

TABLE 2

S. No.	City	Outdoor Design conditions					Temperature at cooler exit. °C	Air quantity to achieve 30°C dbt m3/min/ton	Min. dbt attainable col. 73.3°C	E. T. corresponding to col. 9	Relative Humidity in space	Remarks
		bbt	wbt	rh	ET	Wbd						
1	2	3	4	5	6	7	8	9	10	11	12	13
1.	Agra	41.5	22.0	18	29.7	19.5	24.9	34.7	28.3	*	60	
2.	Ahmedbad	41.5	29.4	37	32.5	12.1	31.2	—	34.5	31.1	—/72	R
3.	Ahmed Nagar	38.8	23.9	29	29.5	14.9	26.1	45.0	29.4		67	
4.	Ajmer	39.4	24.7	30	30.0	14.7	26.9	56.5	30.2	*	69	
5.	Aligarh	40.7	23.1	22.5	29.7	17.6	25.7	40.7	29.0		63	
6.	Allahabad	41.7	25.0	27.0	30.8	16.7	27.5	70.0	30.8	27.8	72/68	R
7.	Ambala	39.9	22.5	22.5	29.5	17.4	25.1	35.7	28.4	*	62	
8.	Amritsar	40.1	23.9	26.5	29.9	16.2	26.3	47.3	29.6	*	65	
9.	Asansol	39.0	25.0	33	30.1	14.0	27.1	60.0	30.4	*	71	
10.	Aurangabad	39.7	25.8	33	30.0	13.9	27.9	83.3	31.2	27.9	75/70	R
11.	Banaras	40.8	26 1	32	31.9	14.7	28.3	100.4	31,6	28.7	77/70	R
12.	Bangalore	32.9	24.7	52	28.1	8.2	26.0	43.8	29.3	*	72	
13.	Bareiily	39.5	25.4	33	30.3	14.1	27.5	70 0	30.8	27.8	75/70	R
14.	Baroda	40.3	29.1	40	31.7	11.2	30.8	—	34.1	31.0	—/74	unsatisfactory
15.	Belgaum	34.0	26.1	55	29.2	7.9	27.3	65.0	30.6	—	76	R
16.	Bhavnagar	40.7	29.1	43	32.8	11.6	30.6	—	33.9	31.0	—/74	unsatisfactory
17.	Bhopal	40.2	23.2	24	29.7	17.0	25.7	40.7	29.0	*	65	—
18.	Bijapur	38.7	27.1	40	30.7	11.6	28.8	146	32.1	29.2	82/73	R
19.	Bikaner	41.7	26.2	30	31.2	15.5	28.5	116	31.8	21.9	76/69	R
20.	Bombay	32.8	26.7	63	29.0	6.1	27.6	70	30.9	27.9	79/76	unsatisfactory
21.	Burdwan	36.7	25.7	42	29.8	11.0	27.3	65	30.6	*	74	R
22.	Calcutta	35.3	27.9	58	30.3	7.4	29.0	175	32.3	29.4	87/78	unratisfactory
23.	Chandlgarh	40.1	23.9	27	29.9	16.2	26.3	47.5	29.6	*	68	
24.	Cochin	30.8	26.2	70	28.1	4.6	27.0	58.5	30 3	27.8	80	unsatisfactory
25.	Coimbatore	34.7	27.5	58	30.0	7.2	28.6	125	31.9	29.0	86/78	unsatisfactory
26.	Cuttak	38.7	29.7	59	32.0	8.8	31.2	—	34.5	31.1	—	unsatisfactory
27.	Dehradun	35.5	21.8	30	27.8	13.7	25.9	42.7	29.2	*	57	

1	2	3	4	5	6	7	8	9	10	11	12	13
28.	Delhi	40.4	23.1	26	29.9	16.5	25.4	48.7	29.7	*	68	
29.	Fatehpur	41.8	25.8	28	30.9	16.0	28.2	97.0	31.5	28.4	75/78	R
30.	Gadag	36.8	27.8	51	30.8	9.0	29.1	195.0	32.4	29.4	88/78	unsatisfactory
31.	Gauhati	30.9	25.4	66	27.6	5.5	26.2	46	29.5	—	76	R
32.	Hyderabad	39.5	26.6	37	30.8	12.9	28.5	116	31.8	28.9	80/72	R
33.	Indore	39.4	25.1	32	29.7	14.3	27.2	62.5	30.5	*	70	
34.	Jaipur	40.8	21.7	18	28.6	19.1	24.6	32.4	27.9	*	58	
35.	Jamnagar	36.4	27.8	52	30.6	8.6	29.1	195	32.4	29.4	88/78	unsatisfactory
36.	Jamshedpur	39.4	28.4	45	32.0	11.0	30.0	—	33.3	30.3	—/75	R
37.	Jodhpur	40.8	25.3	28	30.6	15.5	27.6	73.0	30.9	27.9	74/69	R
38.	Kanpur	41.2	21.5	15	28.6	19.7	24.5	31.8	27.8	*	57	
39.	Lucknow	40.8	25.0	28	30.3	15.8	27.4	67.5	30.7	27.8	72/68	R
40.	Ludhiana	40.1	23.9	27	29.9	16.3	26.3	47.5	29.6	—	67	
41.	Madras	38.5	28.6	48	30	9.9	29.6	440	32.9	29.7	94/88	unsatisfactory
42.	Madurai	37.1	27.2	48	30.6	9.9	28 2	97.5	31.5	28.4	80/72	
43.	Mangalore	32.9	26.7	63	29	6.2	27.6	73.8	30.9	27.8	87/78	unsatisfactory
44.	Mysore	33.3	25.6	54	29.2	7.7	26.7	53	30	—	76	R
45.	Nagpur	42.6	24.5	23	30.6	18.1	27.2	62.5	30.5	*	71	
46.	Nizamabad	41.4	26.1	30	31.1	15.3	28.4	110	31.7	28.7	77/70	R
47.	Nowgong	41.9	24.4	29	30.4	17,5	27	58.4	30.3	*	71	
48.	Patna	37.9	26.7	42	30.5	11.2	28.4	110	31.7	28.7	80/72	R
49.	Poona	36.1	25.6	40	30.5	11.5	27.2	62.5	30.5	*	72	
50.	Puri	32.0	27.8	72	29.5	4.2	28.4	110	31.4	28.7	87/78	unsatisfactory
51.	Raipur	41.8	24.4	24	30.5	17.4	27	58.5	30.3	*	68	
52.	Rajkot	40.5	28.9	42	31.2	11.6	30.5	—	33.8	31.0	—/74	R
53.	Roorkee	39.0	23.4	28	29.5	15.6	25.7	40.7	29.0	*	66	
54.	Salem	37.4	28.6	52	31.2	8.8	29.9	—	29.2	30.3	90/75	unsatisfactory
55.	Surat	36.3	27.6	52	30.5	8.7	28.9	160.0	32.2	29.2	85/75	unsatisfactory
56.	Terpur	30.4	25.0	64	27.8	5.4	—	—	—	—	—	not required
57.	Trichirapalli	38.6	22.6	—	30.2	13.0	27.6	73.0	30.9	27.9	75/70	R
58.	Trivandrum	30.7	29.5	90	29.7	1.2	—	—	—	—	—	not possible
59.	Vizagapatanam	33.3	36.9	62	29.3	6.4	27.8	80.0	31.2	27.9	85/80	unsatisfactory

* Comfort conditions exist. R Relief conditions exist

the air but its temperature is reduced due to loss of sensible heat. This conversion of sensible heat into latent heat goes on till the air saturates. The process is known as direct evaporative cooling,

There are limits to the cooling achieved by adiabatic saturation process. The amount of sensible heat removed can never exceed the latent heat required to saturate air with the water vapour. Thus cooling possibility of air depends upon the relative humidity and wet bulb temperature of the air wet bulb temperature is the lowest temperature that can be achieved by direct evaporative cooling.

Climatic Data for Indian Cities

While designing an air conditioning system it is very essential to study geography and climatic conditions of various regions specially for a vast country like India, where hot and dry climate is a characteristic feature of deserts on one hand and sub zero temperature is a characteristic feature in the Himalayas on other hand. The climatic data, dry bulb temperature (dbt), wet bulb temperature (wbt) and relative humidity (rh) for important Indian cities are given in Table 2. These values correspond to average maximum temperature during the month of May. The data

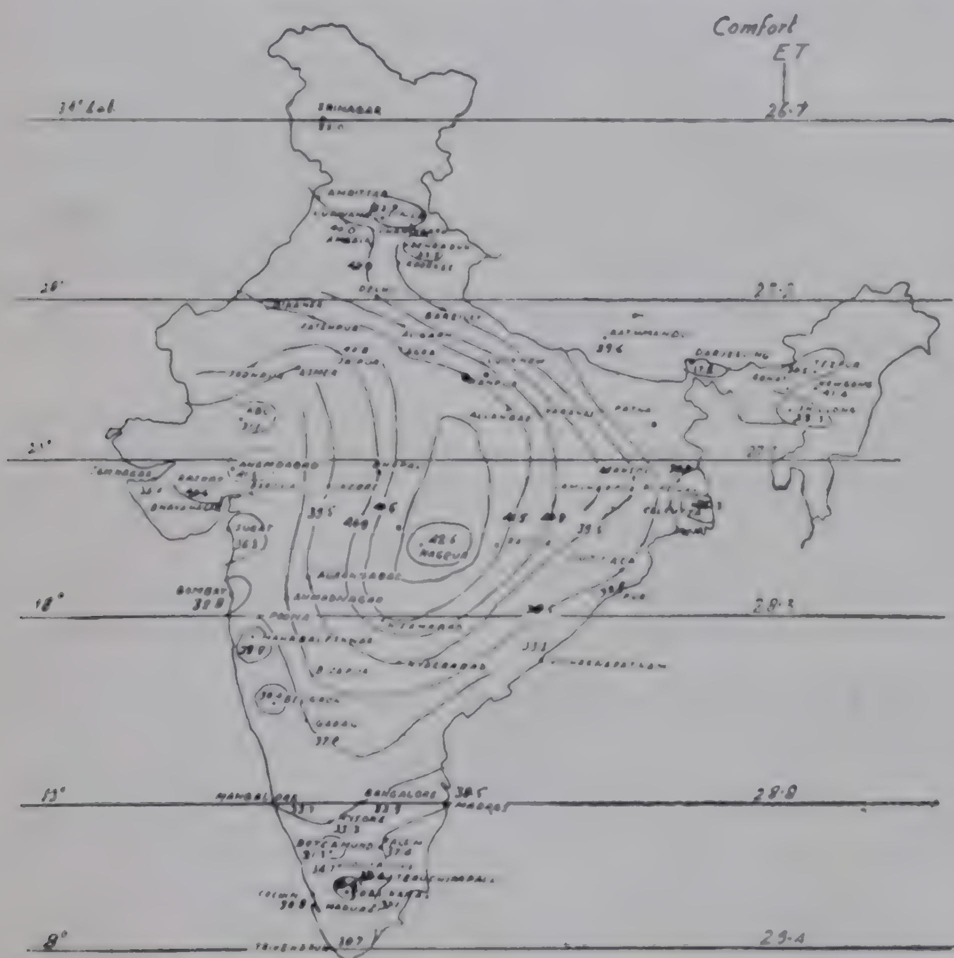

Fig. 1 – Contours of dry bulb temperature

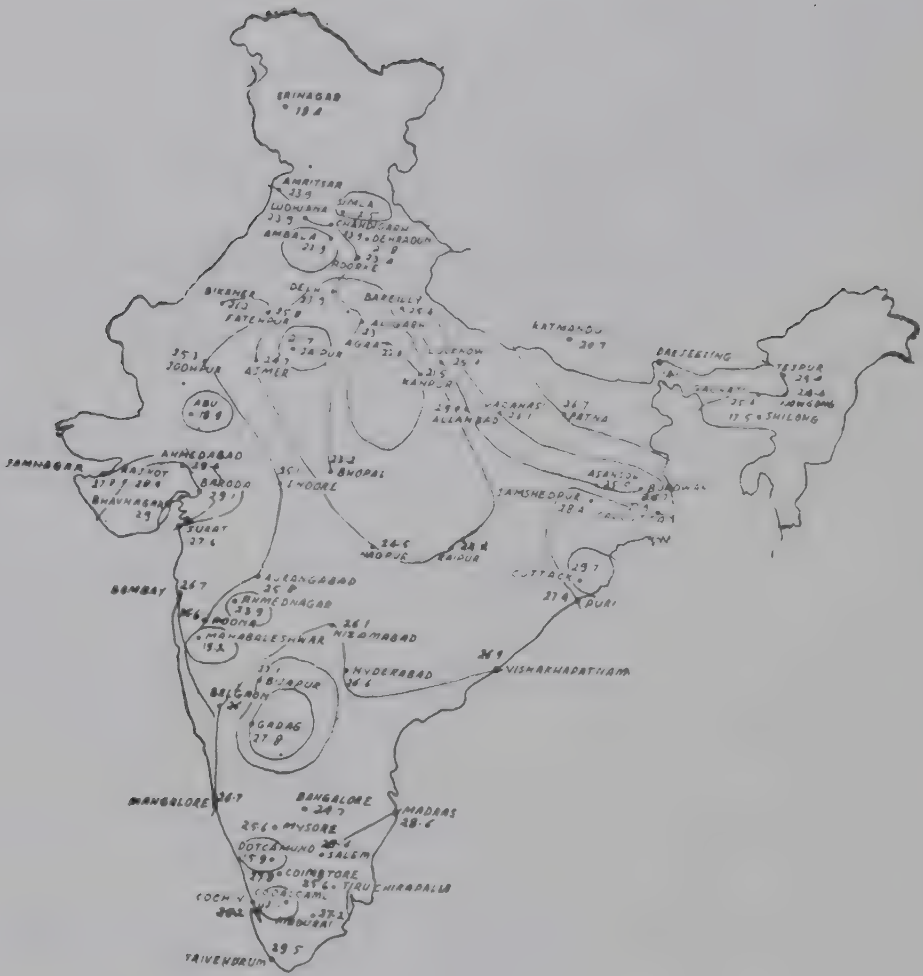

Fig. 2 — Contours of wet bulb temperature.

Fig. 3 – Contours of wet bulb depression.

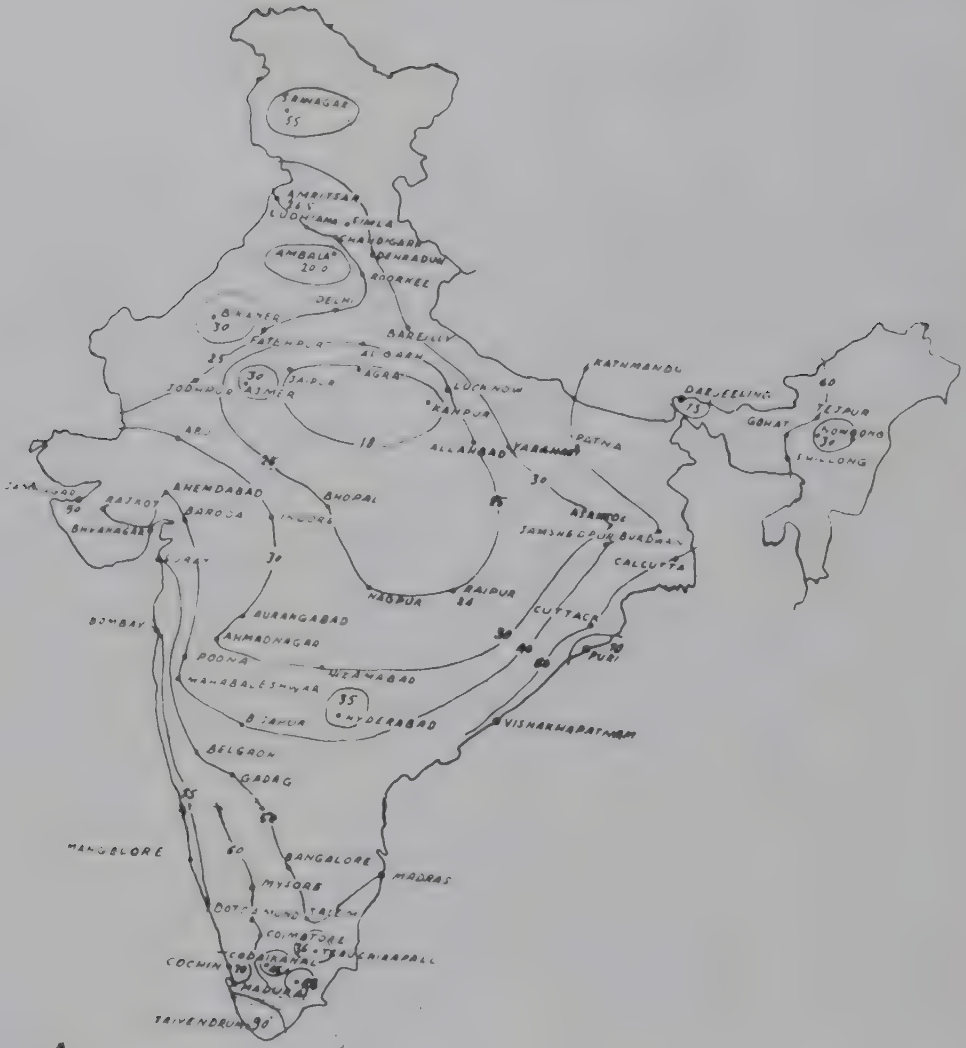

Fig. 4— Contours of relative humidity.

tabulated in columns 2, 3 and 5 have been adopted from reference 5 and corresponding values of relative humidity and wet bulb depression (wbd) have been computed and tabulated in columns 4 and 6 respectively. All these values relate to outside atmosphere and therefore they are termed as outside design conditions. These data have been plotted on the map of India and contours have been drawn. Fig. 1, 2, 3 and 4 show respectively contours of dry bulb temperature, wet bulb temperature, wet bulb depression and relative humidity. Based on these data the authors have made calculations to find out the applicability or otherwise of evaporative cooling to various Indian cities.

Analysis of Climatic Data: Applicability of Evaporative Cooling

Air at outside design conditions flows to the air washer mixes thoroughly with water particles and evaporation of water takes place, cooling down the air. The cooling of air depends upon two factors, the wet bulb depression and saturation efficiency of air washer. Larger is the wet bulb depression, or in other words lower is the relative humidity, higher will be the cooling achieved in air washer. For evaporative cooling Watt[4] has suggested the following limits for wet bulb depression and relative humidity.

1. For comfort—12°C (22°F) or more and rh less than 40%
2. For relief cooling—9.5°C 17°F) or more.

TABLE 1: *Recommended Evaporative Cooling Climate*

Approximate latitude	For comfort cooling			For relief cooling.		
	Max, indoor ET°F	Mini. outdoor design wbd °F	Max. outdoor design wbt°F	Max. indoor ET °F	Mini. outdoor design wbd °F	Maximum outdoor design wbt °F
37–33°N	80	22	77.7	82	17	81
32–28°	81	22	78.7	83	17	82
27–23°	82	22	79.7	84	17	83
22–18°	83	22	80.7	85	17	84
17–13	84	22	81.7	86	17	85
12–8	85	22	82.7	87	17	86

The saturation efficiency of air washer depends upon many factors. It is a function of mass rate of air flow, surface area of water droplets and contact volume i. e. degree of mixing of air and water. It varies between 60 to 95%. For ordinary desert coolers the saturation efficiency may be about 60% or less whereas for capillary air washers it may be upto 95%. For the analysis presented in this paper efficiency of a simple airwasher has been assumed as 85%. With this assumption and considering the wet bulb depression as given in Column 6 of Table 2, the temperature of air at cooler outlet or entry to the conditioned space is given by

$$w = \frac{t_1 - t_2}{t_1 - t^*}$$

OR

$$t_2 = t_1 - w(t_1 - t^*) \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

Where t_1 and t_1 are respectively dry bulb temperatures of air at inlet and outlet of air washer, t^* is the thermodynamic wet bulb temperature and w is the air washer efficiency. $(t_1 - t^*)$ is the wet bulb depression (column 6 of table 2).

As discussed in the article 2 above, for comfort the room conditions in evaporative cooling should be maintained at 30°C and 70% relative humidity, the circulation of air being at a rate of 60 m/min. Knowing the inside design eonditions and temperature of air at cooler exit (from Eq.1), the amount of air required is calculated as follows :

$$\text{Volume of air per min.} = \frac{\text{Heat load per hour in space}}{\text{specific heat of air} \times (t - t_2) \times \text{density of air} \times 60.}$$

Where t is the indoor design dry bulb temperature.

For one ton of cooling i. e. 12000 btu/hr (3000 K cal/hr.), the volume of air in British units is given by :

$$V = \frac{12000}{0.24 \times (t - t_2) \times 0.075 \times 60} \text{ ft}^3\text{/min/ton}$$

$$= \frac{12000}{1.08\ (t - t_2)} \text{ cfm/ton.}$$

In metric units for 3000 KCal/hr. of cooling

$$V = \frac{3000}{0.24\ (t - t_2) \times 1.17 \times 60}$$

$$= \frac{177}{(t - t_2)} \text{ m}^3\text{/min/ton} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

Watt[4] has suggested that the cooled air should gain at least 3.3°C (6°F), in the space before it is discharged. If the temperature rise is less, large quantities of air will be required to cool the space resulting in larger size blowers, ducts, ete. Hence for reasonable economic working of evaporative cooling plants, minimum rise in temperature is assumed as 3.3°C (6°F) and for this rise in temperature the volume flow of air is given by equation (2) as :

$$\text{Vol. } \frac{177}{3.3} = 53.7 \text{ m}^3\text{/min/ton}$$

or

$$\frac{12000}{1.08 \times 6} = 1850 \text{ cfm/ton}$$

Bijlani and Abhat[6] have suggested that the volume of air, for satisfactory evaporative cooling, should not exceed 60m³/min/ton which is agreeable to the values suggested by Watt. Authors have adopted 60 m³/ton/min as the maximum air flow for satisfactory evaporative cooling. The quantities calculated by equation (2) assuming temperature, in the room, of 30°C (Corresponding to ET 26.6°C) are tabulated in column 8. Comparing these values with the value of 60 m³/min/ton, the applicability of evaporative cooling at a particular place can be found out.

Sometimes it is possible that though the wet bulb depression is quite high, the comfort conditions may not be achieved by evaporative cooling due to high wet bulb temperatures. Assuming a temperature rise of 3.3°C in the space, and temperature of 30°C, the dry bulb temperature of air at cooler exit should be 26.7°C. The efficiency of an air washer has been assumed as 85%. Thus the maximum wet bulb temperature at entry to air washer should not exceed 25.5°C (Watt has recommended the maximum wet bulb temperature as 26°C or 78.7°C). Thus we see that even when the wet bulb depression is more than 12°C, evaporative cooling may not give optimum comfort if the wet bulb temperature is higher than 26°C. However, if less than optimum comfort is acceptable, much relief can be obtained if space temperature achieved is say 32°C (instead of 30°C) with a drop in temperature of 7°C (13°F) or more in outside air. Thus for relief cooling, wet bulb temperature 2 to 3°C higher than optimum say 28°C, is permissible. In such cases, from economic considerations, evaporative cooling may be adopted. Three examples are given below to illustrate the applicability or otherwise of evaporative cooling.

Example 1: **Evaporative cooling satisfactory for comfort**

Places where wet bulb depression is more than 12°C and wet bulb temperature less than 16°C e.g., Agra (S. No. 1), wet bulb depression 19.5°C, dry bulb temperature 41.5°C, wet bulb temperature 22°C and outside rh 18% (outside ET 29.7°C).

The condition of air at cooler exit will be $t_2 = 41.5 - 0.85\ (19.5) = 24.9°C$

Volume of air required, $V = \dfrac{177}{30-24.9} = 34.7$ m³/min/ton.

Thus an ET of 26.6°C can be achieved with 24.7 m³/min/ton of air flow and 60 m/min air circulation within the space.

Example 2: **Evaporative cooling satisfactory for relief**

Places where wet bulb depression is quite high but wet bulb temperature is higher than 26°C e.g. Banaras (S. No. 11). Here wet bulb depression is 14.7°C and dry and wet bulb temperatures are respectively 40.8°C and 26.1°C. Relative humidity is 32%, outside ET 31.9°C.

The condition of air at cooler exit will be $t = 40.8 - 0.85 \times 14.7 = 28.3°C$.

Volume of air required, $V = \dfrac{177}{30-28.3} = 100.4$ m³/min/ton.

At Banaras, the amount of air required is much more than the limit (60 m³), optimum comfort cannot be achieved by evaporative cooling except by using a very large fan.

Assuming a minimum rise of 3.3°C, the temperature that can be achieved within the space is 31.6°C at 70% relative humidity corresponding to an ET 28.7°C which can be reduced to about 27.7°C by providing proper air circulation. Volume of air required to achieve such a condition is given by :

$$V = \frac{177}{31.6 - 28.3} = 53.7 \text{ m}^3/\text{min/ton}.$$

This analysis shows that a cooling of 9.2°C has been acquired (with 53.7 m^3/min ton of air) which is substantial. Thus though comfort conditions may not be provided, much relief will be felt.

Example 3 : **Evaporative cooling not satisfactory :**

Places where wet bulb depression is less than 12°C e.g. Madras (S.No. 41), wet bulb depression is 9.9°C, dry bulb temperature 38.5 and wet bulb temperature 38.5 and wet bulb temperature 28.6°C. Relative humidity 48%, outside ET 30.0°C.

Condition of air at cooler exit will be $t_2 = 38.5 - 0.85 \times 9.9 = 29.6$°C.

$$\text{Volume of air required } V = \frac{177}{30 - 29.6} = 440 \text{ m}^3/\text{min/ton}.$$

The analysis shows that a very large fan is required for cooling which is very much uneconomical.

Assuming a temperature rise of 3.3°C. minimum temperature that can be achieved within the space is 32.9°C. This provides little relief.

The above examples show that at places where wet bulb depression is less than 12°C evaporative cooling becomes totally useless. This is the case with almost all coastal towns.

Column 9 shows the minimum temperature that can be achieved by evaporative cooling. This temperature has been calculated by adding 3.3°C to the values of temperature in column 7. However, the objective in evaporative cooling is not to achieve the lowest possible temperature but temperature within the comfortable range i.e. 30°C in the room. ET at places where comfort conditions exist is 26.6°C. At places, where comfort conditions do not exist, but either relief exists or unsatisfactory performance is there, ET corresponding to temperatures in column 9 and 70% relative humidity have been found out and tabulated in column 10.

Conclusion

A study of climatic data for 59 important cities of India has been done. It was found that 17 cities, all of which are at sea coast, have relative humidity above 50%. Evaporative cooling cannot be used at these places. In other 42 cities, evaporative cooling is quite useful. In 22 cities wet bulb depression is higher than 12°C, and wet bulb temperatures lower than 26°C. Optimum comfort conditions can be achieved by evaporative cooling at these places. In rest of the 20 cities, because of wet bulb temperatures being higher than 26°C, evaporative cooling can be used for relief cooling only. To achieve optimum comfort conditions at these places, evaporative cooling may be used in combination with refrigerated air-conditioning.

Fig. 1, 2, 3, and 4 show the various outdoor design conditions. These show the contours plotted by the authors which can be used for design purposes. However, there is scope for improvement in these figures.

India, in its present state of economic development, should try to evolve an intermediate or approximate technology to conserve funds and energy. The paper aims at focussing attention on evaporative cooling, alone or in combination with

air-conditioning, as appropriate technology for India for comfort cooling, wherever possible.

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SESSION VI

Conclusion

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Recommendations

Research on the following areas should be intensified and accelerated with a time-bound programme so that it could be put to beneficial use on a commercial scale in view of the critical food shortage in the country.

Session I

(1) Using irradiation and refrigeration on certain perishable products including their cost structure.

(2) Commercial scale freeze drying of perishable commodities with special reference to fruit juices and evaluation of comparative cost vis-a-vis other techniques.

(3) Feasibility of using individual/instant quick-freezing methods for sea foods, ready-to-serve foods, fruit and vegetable products with liquid nitrogen and different refrigerants.

(4) Intensification of efforts for extending the storage life, prevention of low temperature break-down and maintaining the organoleptic qualities of different tropical fruits with particular reference to mangoes.

(5) Special attention to be given for establishing adequate refrigeration facilities in abattoirs and for the preservation of subsidiary products with special reference to pancreas (required for the manufacture of insulin).

Session II

(6) Further studies and investigations should be carried out to standardise the techniques of heat transfer through heat pipe in cryogenic fields and pulse tube refrigeration.

(7) Computations regarding the losses in refrigeration should be thoroughly investigated in view of the energy crisis.

(8) Further experiments should be continued on freeze drying to understand the basic mechanisms/and to develop various thermo-dynamic equations in view of its urgent use for defence purposes.

(9) Possibilities of reducing the cost of refrigeration in freeze drying should be investigated.

(10) Experimental investigations should be strengthened on studies of vapour-liquid equilibria for refrigerant mixtures.

(11) Various parameters and data required for building enthalpy concentration charts should be studied.

(12) Studies on heat and mass transfer for various tropical food products should be strengthened.

(13) Computer methods should be evolved for standardization and computation of various thermodynamic properties in S.I. units.

Session III

(14) Standard and economic designs for cold stores in various sizes and for various products should be prepared by R & D organisations for the benefit of industry and consumers.

(15) Studies should be made to make multiple utilization of cold stores possible for different products at one and the same time and varying with seasons, in different regions of the country.

(16) Cheap indigenous insulations such as paddy husk should be studied along with the vapour barriers suitable for use with them. For this, it is necessary to establish test facilities at a central place for finding out the permeability of moisture through insulations as well as through vapour barriers.

(17) Design of doors for frozen food stores may be further studied and perfected.

(18) It may be noted that the requirement of a vapour barrier can be waived in jacketted stores since the insulation is generally at a higher temperature than the dew point temperature of the air. Accordingly, cheaper quality insulations can be used in such cold stores.

(19) Existence of one refrigerated sea water plant was reported for fish preservation. Many more such plants and plants of other designs may be tried. When compared to the cost of freezing on board, this method would save the cost of labour required for the pre-cleaning and processing of fish, and would also avoid the thawing that would occur before final processing and packing on shore. The method may also prove useful for storage of fish in processing plants on the shore.

(20) R & D efforts on controlled atmosphere storage should be intensified.

Session IV

(21) Present R & D activities of various universities and laboratories in the field of refrigeration are of vital importance to the industry. There should be close collaboration and exchange of ideas between research and industry for better utilisation of R & D efforts.

(22) The industry should be requested to pose their problems to IIR which in turn may sponsor R & D projects to be taken up by the research organisations.

(23) The computer simulation programme for DX-chillers as outlined could be beneficially utilised by the manufacturers of machinery in order to effect savings of scarce raw materials.

(24) Suitable similar programmes for optimisation of investment may be initiated in research centres for other refrigeration machinery.

(25) Of late, liquid nitrogen freezing is gaining popularity in view of its inherent advantages. Industry should sponsor research and developmental activity in this field.

(26) Refrigerating machinery for cold storages and freezing plants suffer from a very high rate of excise duty due to an erroneous impression that it caters to the affluent section of society. In order that the machinery could be effectively used in the field of preservation of food, by lowering its costs, there is need for drastic reduction of central excise duty.

Session V

(27) Work may be undertaken to precisely determine the effective inside temperature range applicable to Indian conditions for evaporative cooling.

General Recommendations

(28) The Symposium recommends that the NCIIR should establish a liaison with all institutions doing work in the field of refrigeration and air-conditioning and bridge the existing communication gap between institution and institution and institutions and industry.

(29) It is recommended that the NCIIR should take appropriate follow-up action regarding the recommendations made by the Symposium and report the action taken at the time of the next Symposium.

(30) It is recommended that the Proceedings of the Symposium be brought out as a separate publication after editing all the papers presented, along with the recommendations.

(31) It is also recommended that the future organisers of the Symposium device such means as to reduce the number of papers to be presented at the Symposium and give more time for discussions and supply pre-prints of papers as far as possible.

(32) It is recalled that in the 1st National Symposium held at Durgapur in 1972, a recommendation had been made that action should be taken to bring out a Refrigeration Data Book for use in India compiling all available data in the field on the model of the ASHRAE Handbook and Guide. It is also recalled that the ISI had undertaken to bring out this compilation. It is recommended that NCIIR may take suitable action to bring out this publication in consultation with the ISI.

(33) It is recommended that short term courses on specific topics in the field of refrigeration and air-conditioning, cold and freezer storage should be held periodically at institutions where such facilities exist like the CFTRI, CMERI, BARC etc., and NCIIR should take a lead in organising these courses.

(4) It is recommended that to encourage good quality papers being presented, the NCIIR should institute a prize for the best technical paper prepared and presented at such Symposia.

(35) It is recommmeded that in future only metric system of units should be adopted in all papers presented at the national symposia.